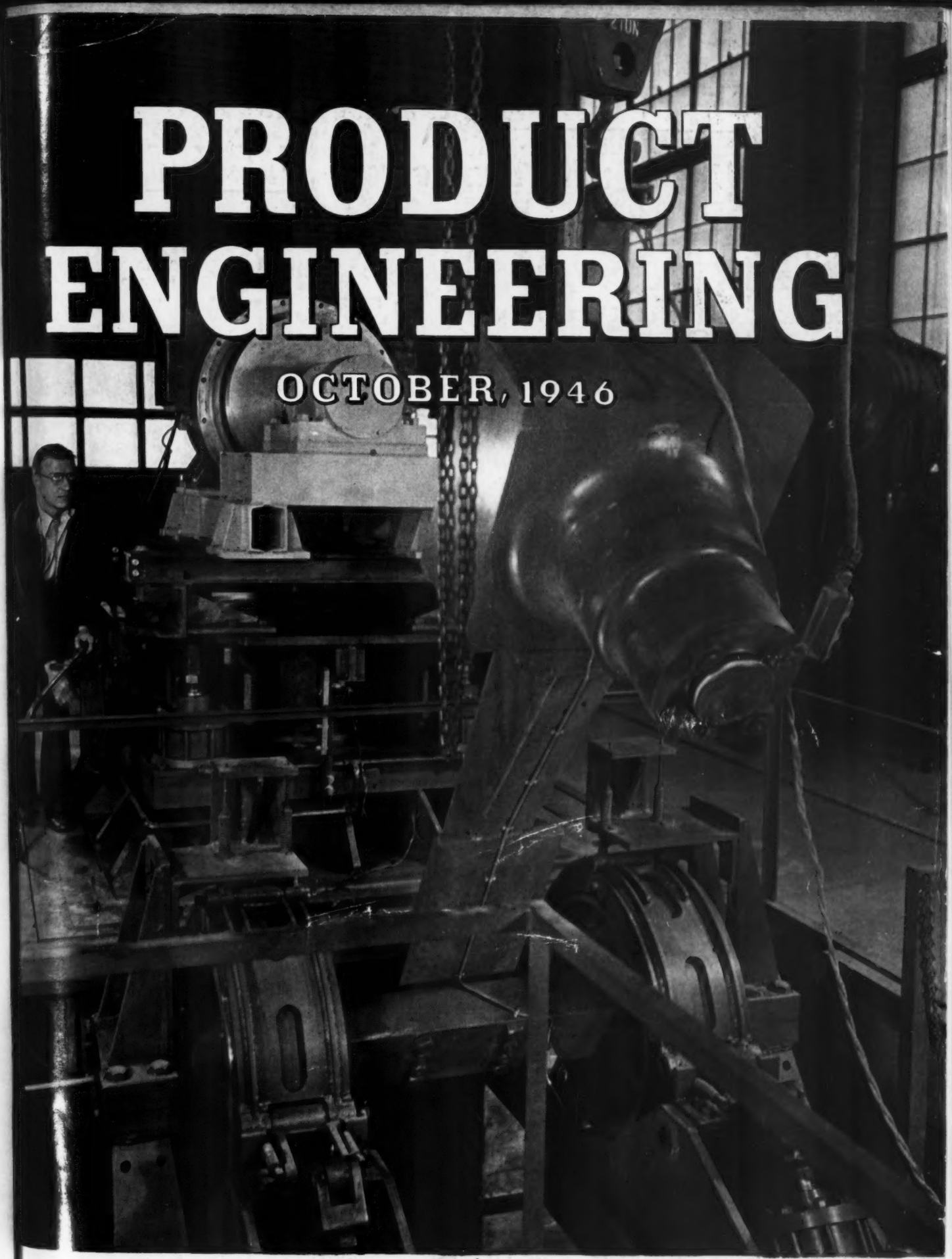
PRODUCT ENGINEERING

OCTOBER, 1946



IN THIS ISSUE—Steels for Cold-Formed Springs + Furnace Brazed Parts + Hydraulic Piping Data + Rotating Shaft Seals + Stresses in Disks + Motor Speed Regulation

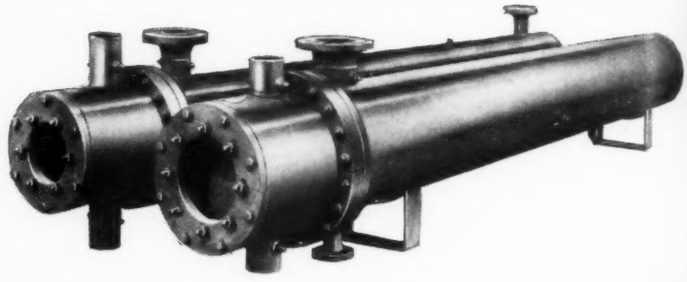
It's the ***FINAL COST*** *that counts*

...keep it low with non-ferrous ***NICKEL*** *alloys*

CUPRO-NICKEL

No Tube Deterioration Indicated after 6 Years of Service

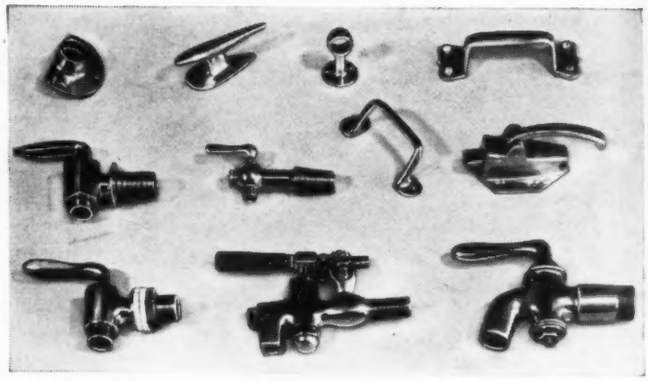
In these feedwater heaters, thin tube walls of .065 gage Cupro-Nickel permit high and rapid heat transfer at working pressures up to 900 pounds p.s.i. Hairpin bend U-tube units of 5/8" O.D. Anaconda Cupro-Nickel Tubes indicate no deterioration after serving 6 years. Characterized by high strength at operating temperatures, along with sturdy resistance to erosion and corrosion, Cupro-Nickel renders economies in applications throughout industry . . . just as it does in these heaters built by Struthers Wells Corp.



NICKEL SILVER

A Metal Combining Beauty with Practical Durability

The beauty of solid Nickel Silver impresses one instantly. Essentially white in color, Nickel Silver possesses strength as well as corrosion-resistant properties superior to the usual brasses. It is a metal that may be cast and worked by simple methods. Available in extruded, rolled, drawn and cast sections, this Nickel alloy lends itself to countless applications. A few items produced by the National Brass Works of Los Angeles, Calif. in 20 percent Nickel Silver, are shown at right.



NICKEL-BRONZE

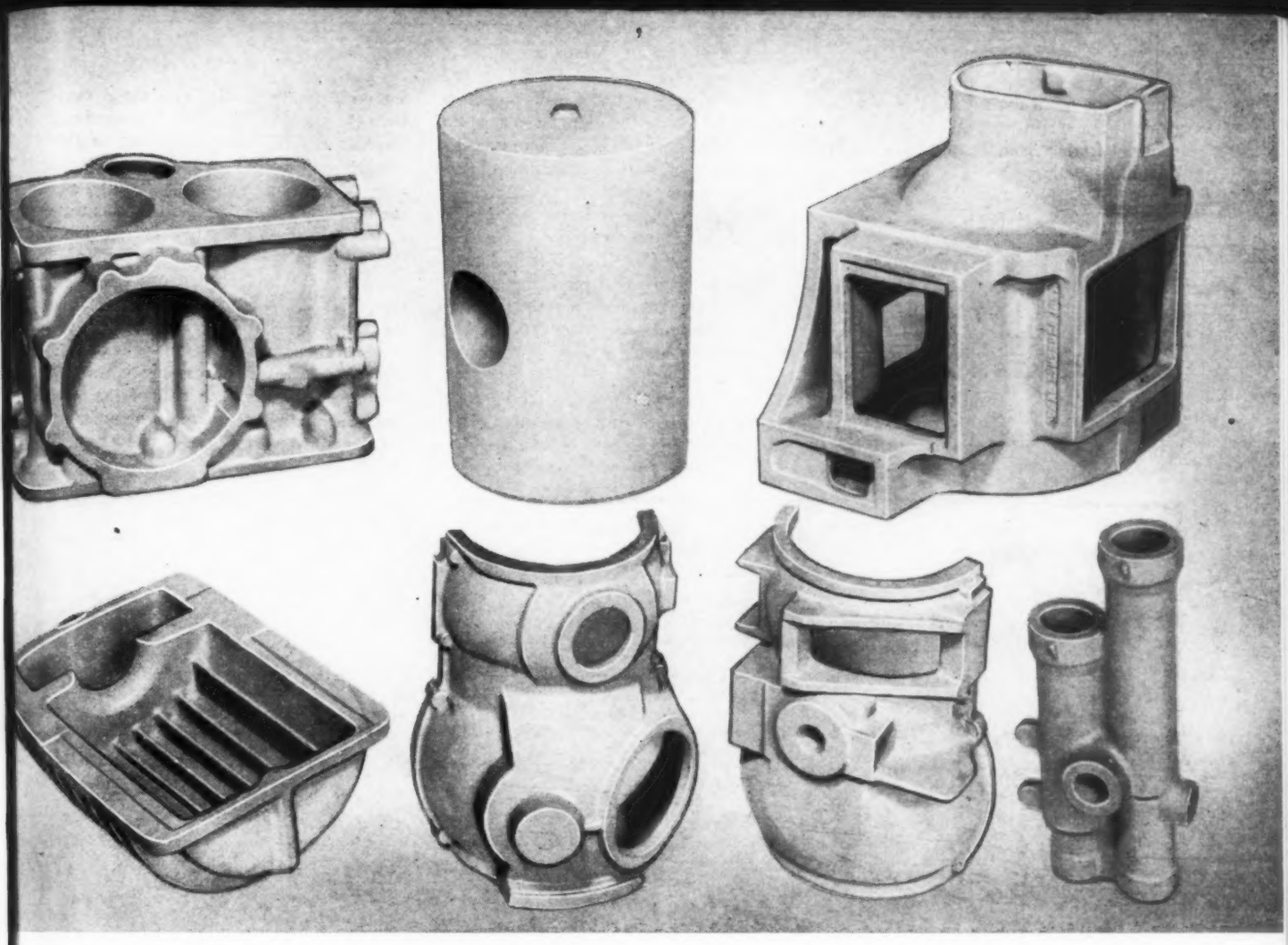
A Little NICKEL Imparts Strength and Toughness

Only 2 percent of Nickel gives these gear rim blanks and transformer case castings what they need. Nickel improves the strength of standard bronzes and is particularly effective in raising elastic properties . . . increasing shock-resistance as much as 25 to 50 percent. Moreover, Nickel increases castability . . . thus it helps to reduce losses due to mis-runs and affords a widened casting range. The photograph shows Nickel-Bronze castings produced by the Philadelphia Bronze & Brass Corp.

EMBLEM OF SERVICE INCO Nickel TRADE MARK

Over the years, International Nickel has accumulated a fund of useful information on the selection, fabrication, treatment and performance of alloys containing Nickel. This information and data are yours for the asking. Write for "List A" of available publications.

THE INTERNATIONAL NICKEL COMPANY, INC. 67 WALL STREET NEW YORK 5, N.Y.



Sand Cast and Permanent-Mold Cast Aluminum Alloy Parts

WALTER J. KLAYER
Aluminum Industries, Inc.

Physical properties of castings made by the two methods, relative costs and production rates are compared. Dimension tolerances, allowable radii, draft and other design data that belong essentially to designed parts, but which are defined by the molds in which the parts are cast, are treated with the view to being compatible with good mold design.

ALUMINUM ALLOY CASTINGS are produced commercially by three principal processes, that is, as sand castings, as die castings and as permanent-mold castings. The physical properties, finish, tolerances, relative cost and fields of application of the castings produced are markedly different.

A sand casting from an inexpensive wood pattern is usually a start to find out if the part can be satisfactorily cast from aluminum alloys. If the castings are large or intricate, the permanent-mold process is sometimes impracticable. Under such circumstances a semipermanent mold, or one composed of metal for its major pieces and of sand cores, often can be utilized.

Any aluminum alloy that will make satisfactory parts by die-casting or sand-casting is usually suitable for permanent-mold castings. Alloys of aluminum with copper, silicon, magnesium, nickel, zinc, manganese and the binary, ternary or quarternary combinations of them in varying percentages are among the most commonly used.

Some alloys are decidedly hot short, or possess peculiar solidification ranges and shrinkages. They must be modified slightly to overcome these difficulties without appreciably altering the basic physical properties, whenever these basic properties are de-

sired in the particular casting that is to be made in a permanent mold or to be changed from a sand casting to a permanent-mold casting.

Physical Properties

Hardness and wear resistance, strength or hot strength, corrosion resistance, thermal conductivity, shock resistance, low expansion, free machining low cost and ease of polishing are generally the properties desired in a material. No one aluminum alloy possesses all of them to the optimum degree for a particular application. A material should be selected for a part on a basis of a suitable balance of properties. Merely to specify aluminum for a part is not specific enough because there is such a wide group of alloys of aluminum from which to make a selection.

To make the proper selection of alloy, detailed information covering a variety of alloys is essential. Table I indicates the wide ranges in physical properties of some of the more common aluminum alloys when cast in permanent molds. The kind of heat-treatment applicable to each alloy is also designated.

Comparative data showing the effects of the the method of casting on some of the physical properties of aluminum test bars are given in Table II. The metal came from the same lot of ingots. The sand-cast test bars were poured at the lowest possible temperature in green sand molds with ample allowance for feeding from gates and risers. The permanent-mold test bars were also cast to insure good feeding, chilling rate, mold temperature and rate of pouring.

Only ultimate tensile strength, elongation, hardness and density are tabulated, the other physical properties being practically in the same proportion. The density of the permanent-mold castings is roughly one percent greater than that of the sand mold casting, which is worthy of note.

Greater strength of the permanent-cold

Table I—Physical Properties of Aluminum Alloys for Permanent-Mold Castings

Constituents Other Than Aluminum, in percent	Heat-Treatment	*Ultimate Tensile Strength, lb. per sq. in.	*Yield Strength, lb. per sq. in.	*Elongation percent	Ultimate Compressive Strength, lb. per sq. in.	Brinell Hardness Number	Shear Strength, lb. per sq. in.	Density, lb. per cu. in. at 70 deg. F.	Expansion Coefficient, Millionths per in. per deg. F.	Remarks
5 Si	None	24,000	10,000	8.0		55	18,000	0.097	13.3	Good corrosion resistance
7 Cu, 3 Si, 1.5 Zn	None	30,000	20,000	1.0	71,000	85	22,000	0.103	12.7	Common casting alloy
10 Cu, 1 Fe, 0.4 Mg	Precipitation only	37,000	35,000	0.5	70,000	115	27,000	0.105	13.0	Excellent wear resistance and high hardness
10 Cu, 1 Fe, 0.4 Mg	Solution and Precipitation	48,000	36,000	0.5	89,000	140	30,000	0.105	13.0	
12 Si, 2.5 Ni, 1.0 Cu, 1.0 Mg	Precipitation only	36,000	28,000	0.5	60,000	100	24,000	0.098	11.0	Low expansion and excellent wear resistance
12 Si, 2.5 Ni, 1 Cu, 1 Mg	Solution and Precipitation	45,000	35,000	1.0	80,000	125		0.098	11.0	
4.5 Cu, 2.5 Si	Solution only	40,000	22,000	10.0	77,000	70	30,000	0.100	13.3	High shock resisting strength
4.5 Cu, 2.5 Si	Solution and Precipitation	45,000	33,000	5.0	82,000	90	32,000	0.100	13.3	High shock resisting strength
4 Mg, 1 Si	None	27,000	12,000	8.0		60	22,000	0.095	14.4	Excellent corrosion resistance
5 Si, 1.5 Cu, 0.5 Mg	Solution only	38,000	23,000	6.0	79,000	70	29,000	0.097	13.0	High shock resistance with good corrosion resistance
5 Si, 1.5 Cu, 0.5 Mg	Solution and Precipitation	43,000	26,000	4.0	83,000	75	30,000	0.097	13.0	High shock resistance with good corrosion resistance

*Note: Average results of 0.505 in. individually cast test bars

Table II—Comparison of Physical Properties of Some Common Aluminum Alloys Cast in Permanent Molds and Green Sand

	Sand Casting				Permanent-Mold Casting				Comparison
Constituents Other Than Aluminum, in percent	Ultimate Tensile Strength, lb. per sq. in.	Elongation, percent	Brinnel Hardness Number	Density, lb. per cu. in. at 70 deg. F.	Ultimate Tensile Strength, lb. per sq. in.	Elongation, percent	Brinnel Hardness Number	Density, lb. per cu. in. at 70 deg. F.	Increase in Tensile Strength, percent
8 Cu	22,000	2.0	65	0.103	30,000	2.0	70	0.104	36.3
5 Si	17,000	6.0	40	0.096	24,000	6.0	50	0.097	26.3
4 Cu, 1 Si, heat-treat No. 1	31,000	7.0	65	0.100	37,000	7.5	70	0.101	12.9
4 Cu, 1 Si, heat-treat No. 2	40,000	4.0	100	0.100	45,000	5.0	110	0.101	12.5
4 Cu, 2 Ni, 1.5 Mg	37,000	0.5	105	0.100	45,000	0.5	115	0.100	21.6
10 Cu, 1 Fe, 0.4 Mg	37,000	1.0	110	0.104	48,000	0.5	140	0.105	29.8
5 Si, 1.25 Cu, 0.5 Mg, heat-treat No. 1	31,000	5.0	60	0.097	38,000	6.0	70	0.099	22.6
5 Si, 1.25 Cu, 0.5 Mg, heat-treat No. 2	35,000	3.0	75	0.097	43,000	4.0	85	0.099	22.8

Note: Each alloy identical in both sand and permanent mold, and pouring governed to lowest temperature that would fill the cavities in each pouring.

Table III—Comparison of Cost Between Sand and Permanent Mold Castings

	Sand Casting	Remarks	Permanent Mold	Remarks
Weight	3.39 lb.		3.01 lb.	12.4 percent lighter
Alloy	$0.15 per lb.	4 Cu, 1 Si	$0.15 per lb.	4 Cu, 2 Si.
Heat-Treatment	$0.05 per lb.	Solution and age	$0.05 per lb.	Solution and age
Casting Equipment	$160.00	1 match plate, 2 on plate 1 core box	$700.00	Complete equipment mold and cores
Castings per Hour	20 average	10 molds per hour includes cope and drag, gate and riser cutting, core setting, closing, pour and shake out 40 cores per hour includes making, setting, baking, touching up and coating.	40 average	Complete castings, single cavity mold.
Cleaning	60 per hour	Gate and riser sawing, trimming and sand blasting (two men)	200 per hour (ave.)	gates and risers sawing (one man)
			200 per hour (ave.)	grinding excess metal (1 man)
Scrap	5.0 percent ave.	Misruns, cracks, shrinks, etc.	2.0 percent average	
Foundry metal loss	5.0 percent	sawing, grindings, splashings, dross, etc.	2.0 percent average	
Cost approximate	$0.50 per lb. or 3.39 x 0.50 = $1.695 ea.		$0.40 per lb. or 3.01 x $1.204 each	

Note: A direct saving of $0.49 each on 1,100 castings amortizes the difference in equipment cost of $540.

casting is the significant physical property. It is accompanied by freedom from sand and dirt, freedom from gas porosity and internal shrinks, smoother surfaces, and closer tolerances as cast.

Costs and Tolerances

An approximate cost comparison on the same part cast in sand and in a permanent mold is given in Table III. This part is a housing for an electric motor. At first the part was cast in sand from a heat-treated alloy of 95 percent Al, 4 percent Cu and 1 percent Si. After producing about 500 parts, which proved that parts made from this alloy were entirely satisfactory, its manufacture was changed to a permanent-mold casting. Not only were the savings in time and cost realized, as indicated in Table III, but also the part weighed less and had a better appearance. Fig. 2 shows the "as cast" part both as a sand casting and as a permanent-mold casting.

Small permanent-mold castings can be held to a tolerance of plus or minus 0.010 in. Castings having one or more dimensions of 6 in. and over preferably should have larger tolerances. A tolerance of plus or minus 0.015 in. can be held on dimensions between 6 and 12 in. Larger dimensions should be given a tolerance of plus or minus 0.020 in. at least.

One percent of the weight is a reasonable weight tolerance on permanent-mold castings. This weight tolerance takes care of the variations in alloy compositions as well as the variations in dimensions. For example, automotive piston castings, which weigh about 1 1/2 lb., have a weight tolerance of 1/4 oz. Automotive pistons usually are cast with a small balance pad having a total volume equal to the weight tolerance on the rough casting. This pad can be machined during the balancing operation to give each piston a proper final weight.

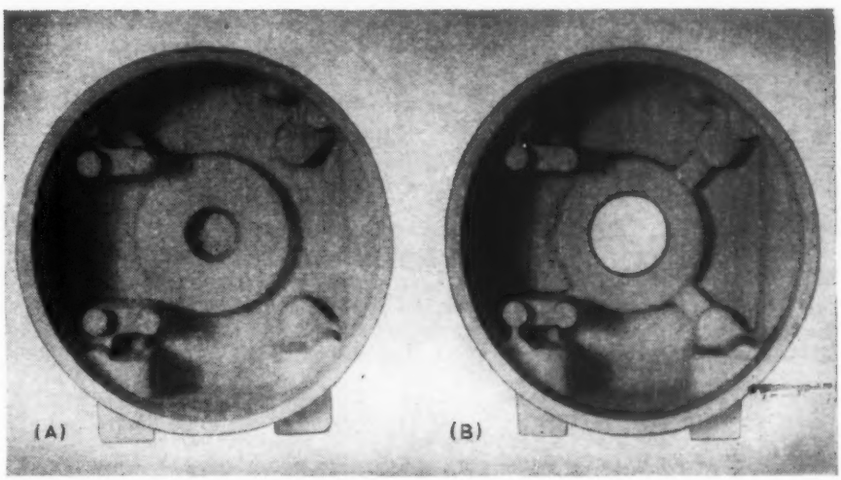


Fig. 2—(A) Sand-cast motor housing weighs 4 lb. 6 oz. (B) The same part as a permanent-mold casting weighs 3 lb. 10 oz. Compare appearance of surfaces.

Weight tolerances of less than two percent cannot be maintained on sand castings. The following sand casting tolerances have wide application and are offered as a guide to the designer when specifying dimensions on a drawing: A tolerance of plus or minus 0.020 in. for dimensions under 6 in., plus or minus 0.030 in. for dimensions between 6 and 12 in. and plus or minus 0.040 in. for dimensions greater than 12 inches.

With no intention toward defining limits of sizes, shapes and weights of cast aluminum parts a few parts are mentioned here as common examples. Small parts, such as brake pistons weighing an ounce or two, piston pin retainers and plugs measuring less than an inch in diameter, can be cast to compete in cost with other methods of manufacture. Wash machine agitators, stove grills, shampoo boards, refrigerator parts, meat cutters, vending machine parts, cooking utensils, steering gear and shock absorber parts, bus and street car windows and a multitude of similar articles, both uniform and irregular in shape, are ideal for production as aluminum castings. Diesel pistons weighing 20 lb. or more, cylinder heads for internal combustion engines, motor and machine bases are examples of larger and heavier pieces.

Processes

In the permanent-mold process, the molten alloy is poured into a previously heated and assembled cast-iron mold. As soon as the alloy solidifies, the core pieces are taken out, the mold parted, and the casting removed. The mold halves are

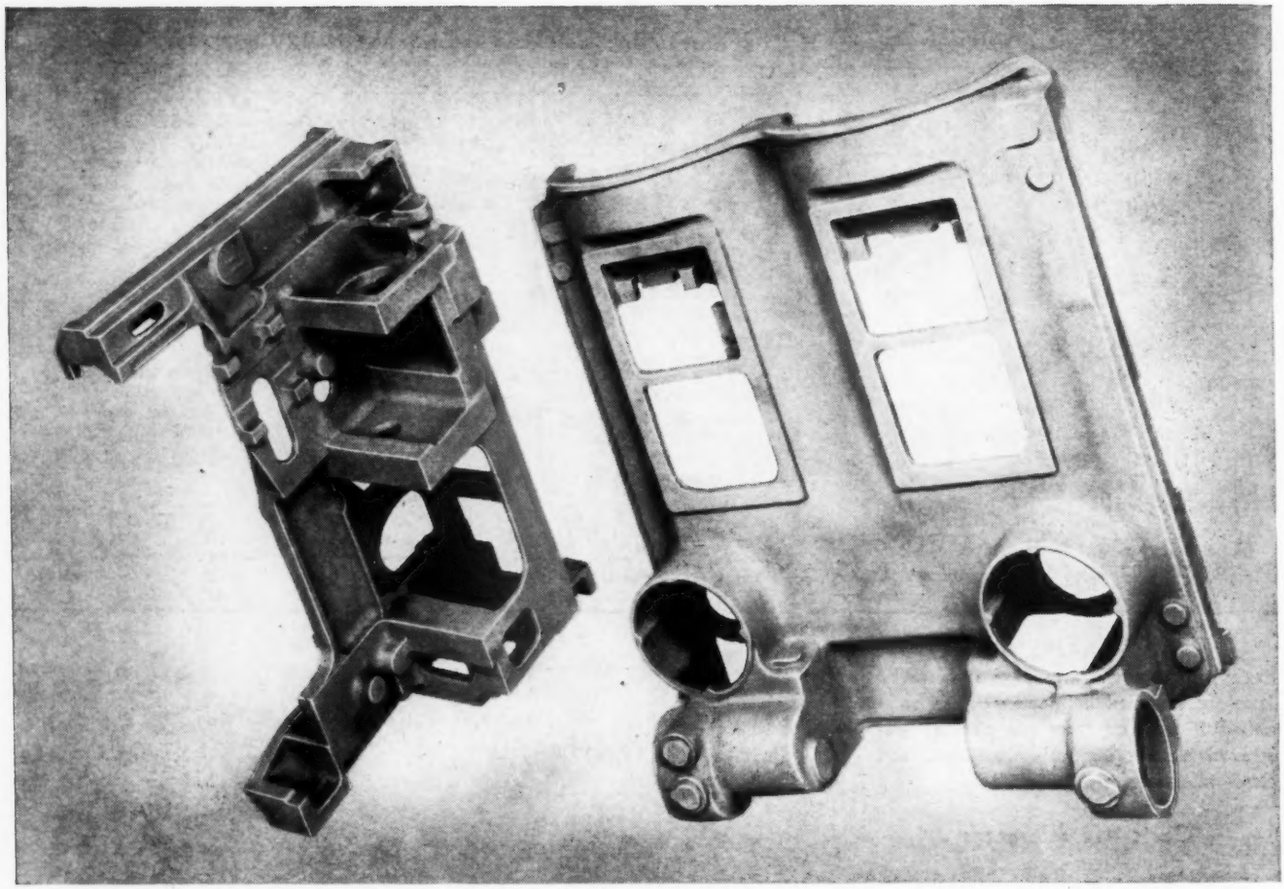


Fig. 3—Sand castings weighing 49 (left) and 32 lb. Green-sand cores, ram-up cores and set cores were used in making these large castings.

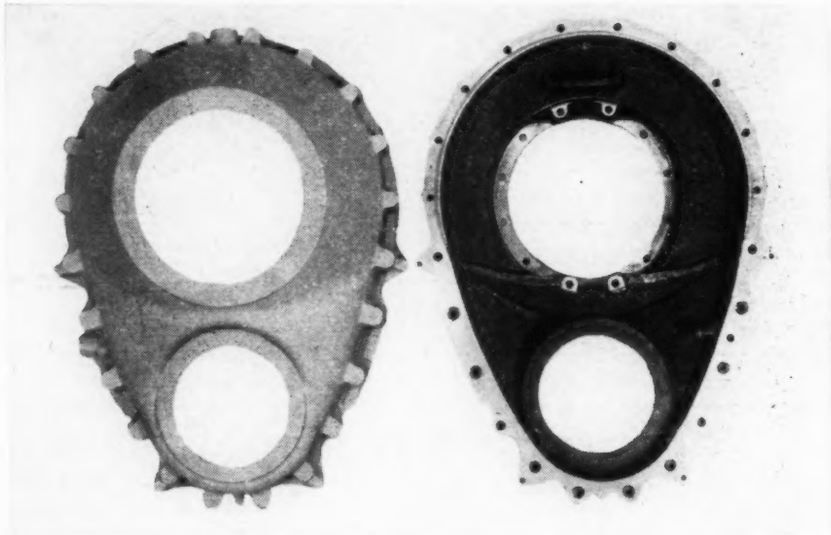


Fig. 4 — Gear-case cover weighing 13 lb., cast in a semipermanent mold.

again brought together, the core pieces inserted, the mold locked, and another pour is made. In production, this procedure is repeated over and over again. Permanent molds that have made over 100,000 parts per cavity in a multiple cavity mold, with only minor repairs to the mold itself, are not unusual.

The sand casting method is applicable to the most varied types of parts as regards size, shape, intricate coring and other features. The production of castings in sand consists in pouring the molten alloy into a previously prepared sand mold, the alloy assuming the shape of the mold cavity, and retaining that shape, approximately, when solidified. In making the mold, sand of the required texture and moisture content is rammed into a flask using a pattern, or match plate, to give the required contour of the casting. The cores are made of sand and are set in the molds. After the casting is poured, the entire mold is destroyed in order to remove the casting and shake out the core.

Designers in comparing costs of molds for the method of casting should not overlook the matter of mold maintenance. Some aluminum foundries keep permanent molds in repair at no cost to the owner. Patterns and core boxes for sand molds on the other hand are repaired at the owner's expense. One foundry will store molds indefinitely, without charge; another will charge a retainer fee and may dispose of the mold if the owner neither reorders castings nor removes it within a reasonable period. These and other incidental costs properly should be included in a cost estimate and comparison.

Permanent Mold Design

Normally the permanent mold is designed to have an annealed, grey iron body, which takes the place of the cope and drag in the sand process, and grey iron cores. Bodies of collapsible or intri-

cate permanent molds sometimes are made from a high carbon or alloy steel that holds its hardness and shape well at 800 deg. F. Permanent molds consist of two or more parts. The cavity is machined either in adjoining parts or entirely in one part, depending upon the casting and other design factors. The cores, if necessary, are machined to size and shape so as to give the desired contour to the inside of the casting and provide the required wall thickness and intricate inside shapes. Push out pins and core pins are made from alloy steel, hardened to resist wear. Base plates are usually grey iron or cast steel.

Correct mold design is an important factor in the successful production of permanent-mold castings. Its design must produce a quality casting within the tolerances specified, yet its design is closely related to the method of gating, the type, shape, weight and thickness of mold to be used. Venting and gating are problems that must be analyzed from a metallurgical aspect as well as from a production and engineering standpoint. They are more the problems of production engineers than of designers.

Shrinkage of the alloy must be accurately known so that the mold can be designed to proper size to yield the finished casting dimensions specified. In this matter the part designer can be helpful materially by a full and accurate specification of the alloy from which the part is to be cast. Solidification range, pouring temperature, liquid and solid shrinkage, heat capacity, and fluidity, though not of great concern to the part designer, should be allowed for by the mold designer.

Rarely can a mold that is designed for one alloy be used to provide parts for another alloy without returning the mold to the die shop for alteration to suit shrinkage of the other alloy. For instance, a 10 percent copper alloy shrinks 0.0070 in. per in. and a 13 percent silicon alloy shrinks only 0.0025 in. per in. The mold designer will find shrinkage values helpful when considering alloys for substitution or in specifying alternate materials on mold drawings.

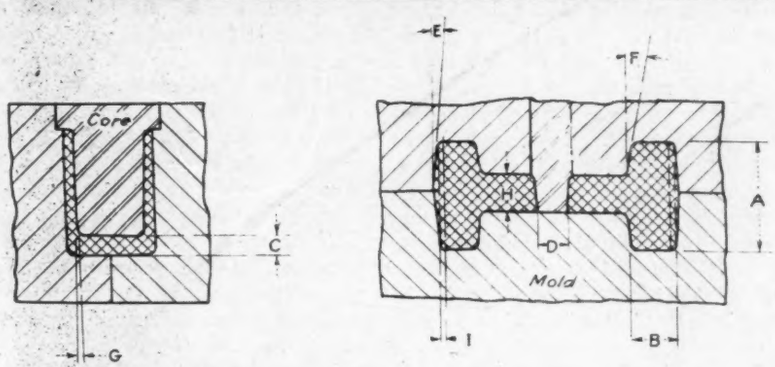
The gate and risers must be designed to suit each casting from the selected alloy to assure soundness, freedom from internal shrinkage and draws, and ample size to insure filling of mold cavity. Sections of poorly designed castings may be impossible to feed. Heavy sections adjacent to thin sections may cause chilling of the metal during pouring. These are faults in the design of the parts and require correction on the drawing board.

Gates must be large enough that the entire mold cavity will fill before the metal drops below the solidification temperature. The alloys have different solidification temperatures, but a pouring temperature of 150 to 200 deg. F. above each solidification temperature is usually ample. Gates also influence production rate, especially under continuous operations where a predetermined number of pieces per hour is necessary to insure satisfactory castings from every standpoint. Moreover, with proper gate design, the operator can only feed metal from the ladle as fast as traps or restricted areas will allow metal to flow through them and also the volume of metal in the shrink pads will keep the mold at a proper operating temperature with no aid from external heating.

A mold wash is indispensable. Vents are necessary to allow entrapped air to escape from the mold. The bauxite and whiting washes act as insulators; the graphite washes act as lubricants; and the talc and graphite washes act as conductors.

Simple and satisfactory mold design depends much on how the part to be cast is designed. Thin sections, adjacent thin and thick sections, sharp fillets, reentrant angles, tolerances, truly flat surfaces, reverse draft or proper allowable draft tend to make mold design easy or difficult. Table IV provides mold tolerances that can be applied in the design of parts.

Table IV—Permanent Mold and Sand Casting Tolerances



Dimension (see diagram)	Permanent Mold	Sand
A, Across parting lines	± 0.010 in. minimum (a)	± 0.030 in.
B, Between points produced by one part of mold	± 0.010 in.	± 0.030 in.
C, Between points produced by core and mold	± 0.010 in. minimum (a)	± 0.030 in.
D, Diameter of small cores	¼ in. minimum (b)	¼ in. minimum
E, Outside draft	1 deg. minimum, 3 deg. desired	1 deg. minimum, 3 deg. desired
F, Draft in recesses	2 deg. minimum, 4 deg. desired	2 deg. minimum, 5 deg. desired
G, Draft on cores	2 deg. minimum	none
H, Web thickness	⅛ in. minimum	3/16 in. minimum
I, Finish allowed for machining	1/32 in. to 1/16 in.	1/16 in. to ¼ in.

Note: (a) ± 0.0015 in. per in. additional above 6 in.
(b) Depth to diameter ratio at least 3 times.

Advantages of Permanent-Mold Process

Demand for aluminum-alloy castings made in permanent molds is growing rapidly. There are several reasons for the greater demand. This method of casting, though essentially a quantity production process, is competitive in cost with sand castings when quantities of parts exceed 500. The rate of production in permanent molds is more rapid than in sand castings, and low-priced operating labor can be used for pouring the castings. In comparison with die casting the mold costs are materially less than die costs in die casting.

Skillful foundry practice, especially in molding technique, is highly important in sand founding. This is not true in permanent-mold work where mold operators can be taught the essentials of the process in a few hours and can be producing satisfactory castings at fair speed in a few days, assuming the mold has been properly designed so that the human element is negligible.

Permanent-mold castings rank higher in mechanical properties, because of fine grain size, soundness, and excellent physical properties. They have greater strength and hardness, higher elongation, greater resistance to impact and alternating-fatigue stresses, and greater resistance to corrosion than sand castings of the same alloy. Permanent-mold castings machine well, with less machining scrap than sand castings. The fine grain also improves the polishing and finishing qualities of the castings.

Valves, Piping and Design Data For Control of Hydraulic Systems

V. M. WELTZIEN
The Sinclair-Collins Valve Company and Valvair Corporation

Variations in the kinds of control valves and the service for which each is suited are discussed. The uses of pilot valves and cycle controllers are illustrated by piping arrangements from typical installations, accompaned by descriptions of the functional design and performance.

AUTOMATIC and semiautomatic control of machines and equipment through the mediums of air, steam and other fluids are increasing in demand. Hydraulic presses for the manufacture of products from rubber and plastics, presses for blanking, coining, swaging, stamping and other purposes, and many machine tools are hydraulically operated by control valves, which in turn are governed by pilot valves and electrical devices.

Although the variety of piping arrangements and valve combinations is endless and new conditions and requirements are always being confronted, the design of such a control system generally can be made a simple procedure. It consists mainly of selecting from the devices now on the market units of suitable type, size and responsiveness to changes in flow and pressure, of connecting the units together with piping, and finally of proving the design by comparison with a satisfactory installation actually in service or by a test installation of the new design.

Foremost in the design of a control system is the specification of the sequence of operations during one cycle. Means for starting the cycle and of resetting the system for the next cycle are other important matters for consideration. Usually they determine whether the control system is to be classed as automatic, semiautomatic or manual.

Types of Control Valves

Diaphragm control valves can be the basic units of the control system. Figs. 1, 2 and 3 illustrate some common types. Functionally, a control valve consists of a valve piston attached to a diaphragm. The axial position of the valve piston in the valve body controls the flow of hydraulic fluid to a press ram, or other processing equipment. Valve piston position is determined by differences in pressure, usually air, on opposite sides of the diaphragm. Generally springs also balance the air loading on the diaphragm.

A direct acting valve, Fig. 2, is one that requires air pressure on the diaphragm to shut the piston against the inlet pressure; conversely a reverse acting valve is one that requires air pressure on the diaphragm to open the piston to admit flow of the hydraulic fluid. Note is made that this usage of direct and reverse acting is not universal with valve manufacturers.

If different axial positions of the valve piston cause hydraulic fluid to flow first to the press ram and then to drain from it, the control valve is termed a 3-way valve. Fig. 3 illustrates a valve in which the pressures exerted by the hydraulic fluid are equalized so as to cause no resultant force along the piston axis. This type is termed a fully balanced valve. About the same air load is required on the diaphragm to close and open the valve regardless of the hydraulic fluid pressure.

The 2-pressure valve, Fig. 1, functions to cause flow to the press ram of hydraulic fluid first at one pressure (low) and then at another or high pressure. With air pressure exerted on the diaphragm or top, the valve piston is down and the lower valve seat is open. Low pressure hydraulic fluid in chamber *J* opens check valve *K*, allowing flow of hydraulic fluid to the ram. As soon as the pressure builds up to some predetermined point, such as 150 lb. per sq. in., the plunger *P* is lifted and high pressure hydraulic fluid flows to the ram.

When this happens, check valve *K* closes, thus preventing high pressure fluid from entering the low pressure line. High pressure may be held as long as desired merely by retaining the air pressure on the diaphragm. Upon release of the air pressure, the lower valve seat closes, the upper valve seat opens and plunger *P* falls to stop the flow of high pressure fluid. The hydraulic fluid drains as the ram is lowered. The 2-pressure valve is used in a variety of hydraulic control systems.

These valves are designed to operate on pressures as low as 75 lb. per sq. in. and on high pressures up to 4,000 lb. per sq. in. They are hydraulically balanced. Other hydraulic, diaphragm-actuated valves are available in sizes ranging from 1/4 to 4 in. to operate on pressures from zero to 4,000 lb. per sq. in. Valves are built for both hot and cold service. The temperature of the liquid flowing through the valve, as well as the pressure and quantity of fluid to be handled, should be taken into consideration in designing a control system.

Selecting the size of a control valve can be based on a calculated rate-of-flow of hydraulic fluid through it in gal. per min. But this is rarely done because resistance to flow of the piping, fittings and other elements of the system, as well as the resistance through the valve itself, are uncertain and variable quantities. If flow sizes are computed, a safe rule to follow is to assume that from one-fourth to one-third the total pressure drop of the system occurs in the control valve. Normally the maximum flow through the valve should not exceed twice its rated capacity.

As a rule control-valve sizes will be proper if made equivalent to the hydraulic inlets on purchased equipment, such as hydraulic presses. The traverse area of the valve usually is about equal to the traverse area of the pipe. In the event that the valve size determined in this manner turns out to be inadequate, then pipe size as well as valve size should be increased. Sometimes changes to simplify the piping arrangement will correct the deficiency; increasing the valve size alone seldom does.

Safety requirements and the matter of whether the operation of the system is to be automatic, semiautomatic or manual also should not be overlooked in choosing a control valve. A common practice has been to submit control problems to the engineers of valve manufacturers, who design the control system and make a piping diagram in addition to recommending valves suitable for a given installation. This design service is always done free of charge.

Pilot valves comprise one group of devices for energizing control valves with compressed air. Some pilot valves of ingenious design have been developed within

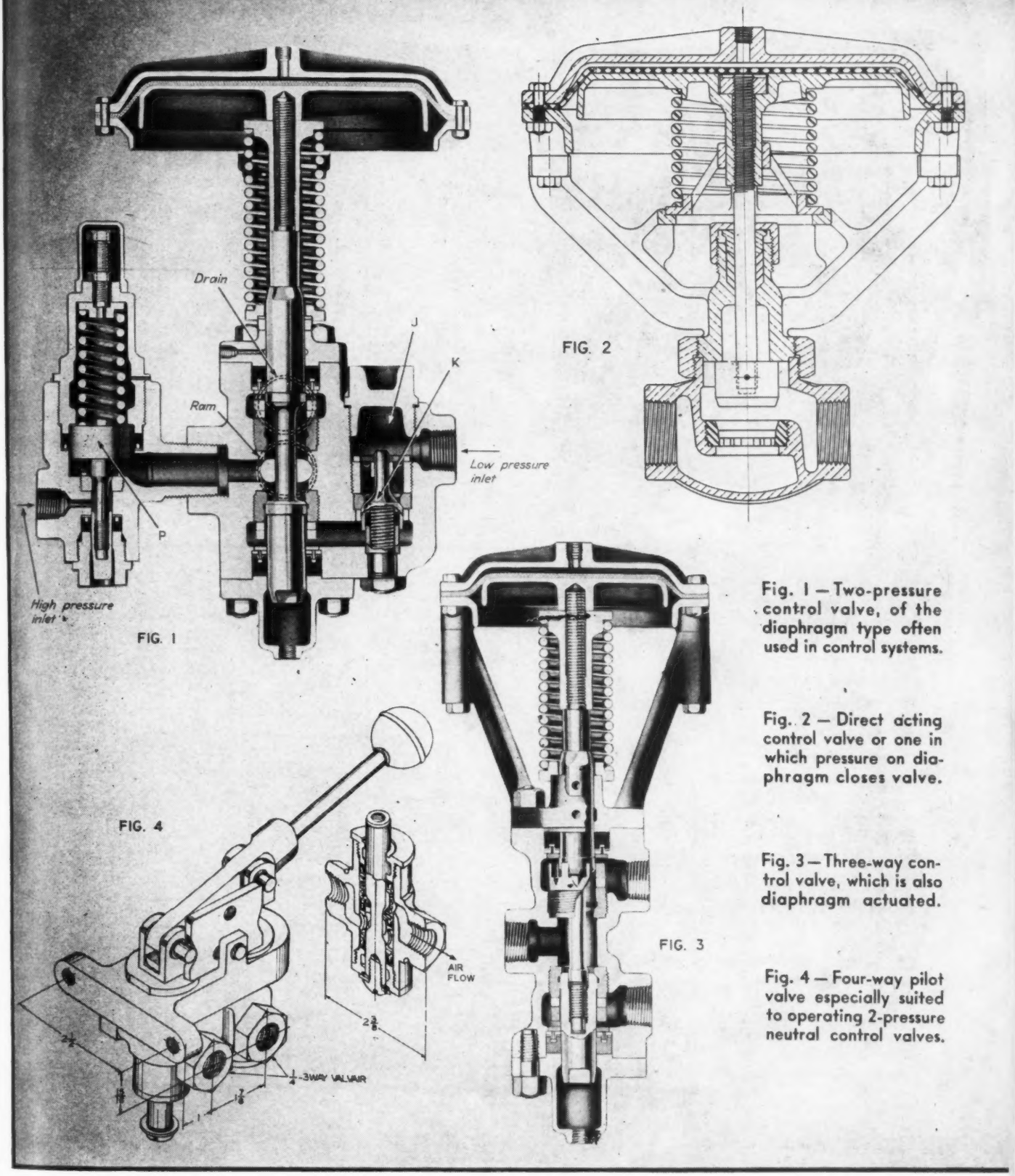


Fig. 1 — Two-pressure control valve, of the diaphragm type often used in control systems.

Fig. 2 — Direct acting control valve or one in which pressure on diaphragm closes valve.

Fig. 3 — Three-way control valve, which is also diaphragm actuated.

Fig. 4 — Four-way pilot valve especially suited to operating 2-pressure neutral control valves.

the last year, only one or two being required to energize complicated systems of control valves.

Two-, three- and four-way pilot valves are available. These can be knob, lever, foot, cam, treadle, diaphragm and solenoid actuated. Fig. 4 shows a lever operated pilot valve suitable for governing two-pressure control valves. Sizes of pilot valves range from 1/4 to 1 in. pipe size and can be used on air lines having pressures from zero to 200 lb. per sq. in. This range of sizes and pressures allows wide latitude in selecting pilot valves suited to any system.

Electrically driven cycle controllers are another group of devices for energizing control valves. Fig. 5 is typical of this type of controller, which has three principal parts: A camshaft that actuates the pilot valves or switches; a timing unit; and a reset mechanism, each part being driven by a separate motor.

Control System Design

An arrangement of units into a simple system for controlling the operation of a blanking press is shown in Fig. 6. The lever-operated, 1/4 in. pilot valve, No. 1,

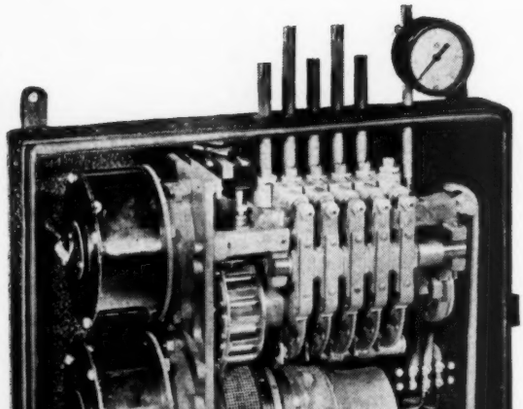


Fig. 5 — Electrically-driven cycle controllers through pilot valves and switches can perform automatically a sequence of operations such as are required in the forming and curing of rubber tires.

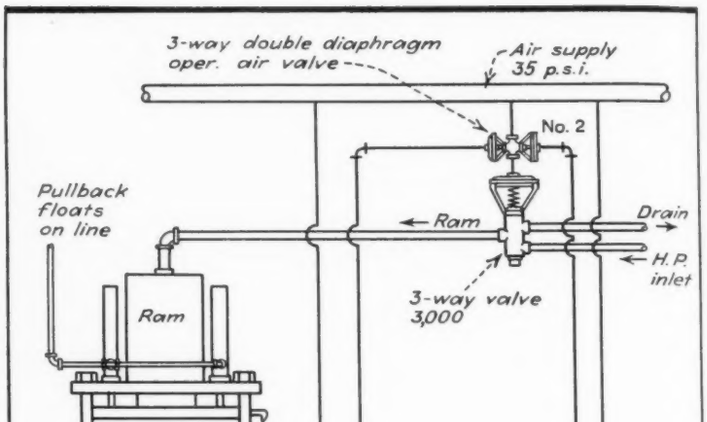


FIG. 6

Fig. 6—One control valve with pilot valves to energize it comprise a satisfactory system for controlling press-ram movements.

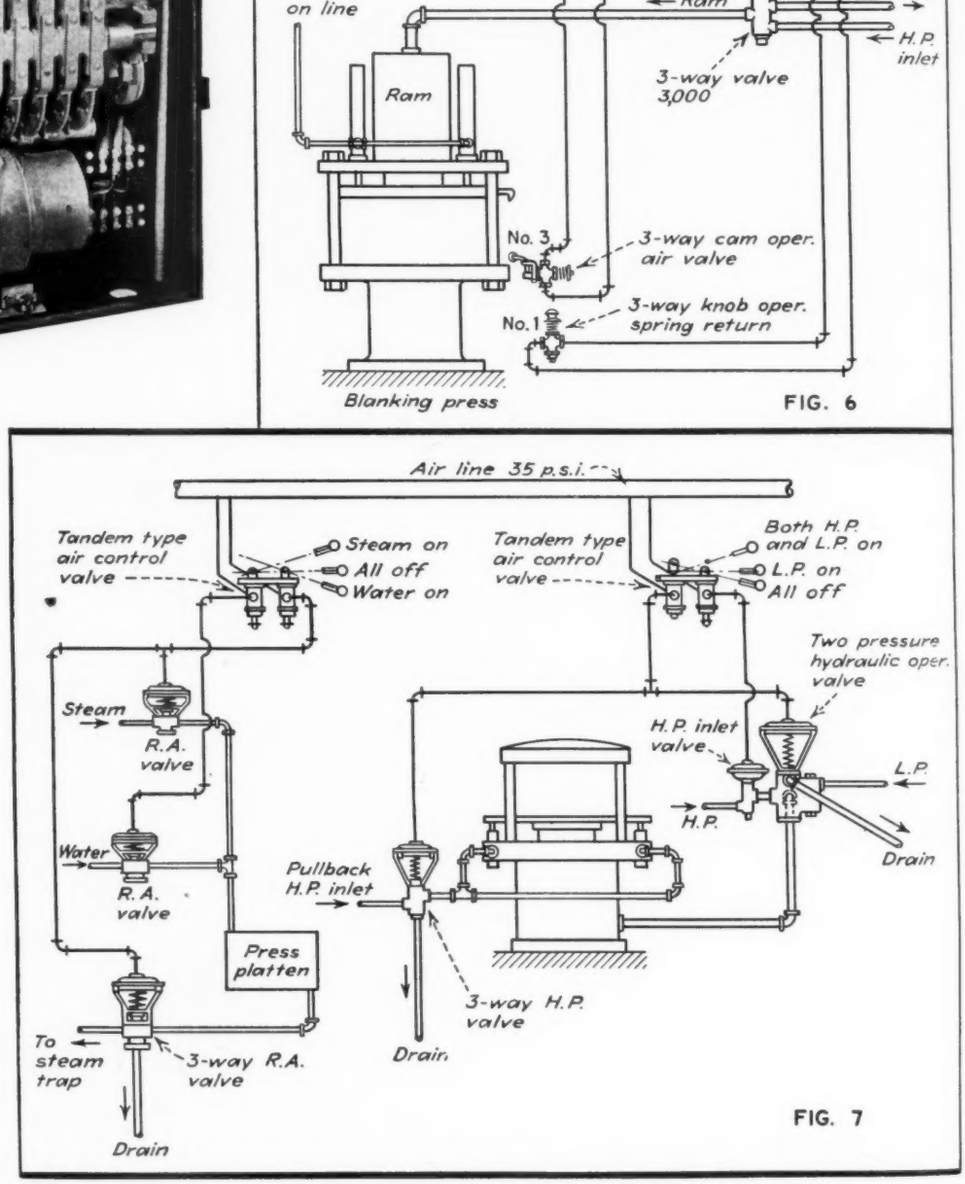


FIG. 7

Fig. 7—A semiautomatic control system such as this is used extensively in hydraulic presses for the manufacture of tires and other rubber products.

starts the motion by opening the 1/2 in. double-diaphragm pilot valve, No. 2. The latter energizes the control valve in the 3-in. hydraulic line. As the ram moves downward to do work it trips a cam-operated pilot valve, No. 3, that closes pilot valve No. 2. This releases the air loading on the diaphragm of the control valve. The control valve opens to drain the ram and the pullback returns the ram to position for the next cycle.

In the manufacture of rubber products the press platen needs to be heated to cure the rubber. Fig. 7 is an arrangement whereby complete control of the system is centered in two pilot valves. The pilot valve on the right operates the ram and pullbacks while the pilot valve on the left controls the heating, cooling and draining of the press platen.

Cycle controllers are widely used on rubber presses for the reason that a cycle controller can be set completely to take over the operation of a press. It will function to close the press on low pressure and cause high pressure to cut in automatically; to bump or breathe the mold; to hold the press on high pressure for any desired time from a few seconds to 8 hours; and to lower the press for reloading. The controller also can be set to cause steam and water to flow to the platen, to operate pullbacks and to hold the ram in any predetermined position for loading, unloading or pulling mold pins. Fig. 8 is an arrangement containing a cycle controller, which provides automatic control and operation of the system in the manner just described.

Certain intricate molding jobs, which occur primarily in the manufacture of plastic products, require precision control of the ram. To meet these requirements a 2-pressure, 5-diaphragm control valve, with throttling control, was designed. Fig.

9 shows this 5-top neutral valve arranged for operating a standard press. As shown it is energized by pilot valves, but a cycle controller, or a cycle controller and pilot valves, could be arranged for the same use.

If the pilot valve is thrown to position "ram-up" air flows to diaphragm 1 to close the drain and to diaphragm 2 to allow low pressure hydraulic fluid to flow to the press at full capacity of the valve. The ram moves upward at full speed. If the ram must be stopped during its upward movement, the pilot valve is shifted to position "neutral." Air is exhausted from diaphragm 2 and the low pressure inlet is closed. The drain does not open because diaphragm 1 remains energized.

At positions in the upward movement of the ram where its speed must be reduced, a cam-operated pilot valve admits air to diaphragm 4, thereby throttling the low-pressure hydraulic fluid. The amount of this throttling is determined by an adjustment that limits the travel of the valve piston when energized from the diaphragm 4 end.

The ram can be reversed during upward travel by shifting the pilot valve to the position "ram down." This exhausts the air from both diaphragms 1 and 2. The one opens the drain to the ram and the other closes the low pressure inlet.

Another lever-operated pilot valve is used to admit high-pressure hydraulic fluid. The high pressure control is used with a 3-way pilot valve so as to interlock the drain side of the neutral valve. This prevents high pressure being turned on while the system is open to drain.

The cam-operated pilot valve causes throttling during downward movement of the ram as well as during upward travel. Through diaphragm 3 it partially closes the valve opening to drain. Just as throttling was used to reduce the rate of upward movement so throttling can be applied to slow down decent of the ram. This time the adjustment is made to limit the movement of the piston attached to diaphragm 3. Once the speed is fixed no further adjustment is required.

Pilot and control valves can be arranged and fitted with safety mechanisms so as to prevent movement while the press is being loaded or if the operator makes a mistake in the order of operations. The 3-way control valve, Fig. 8, on the high pressure line illustrates a good design for safety.

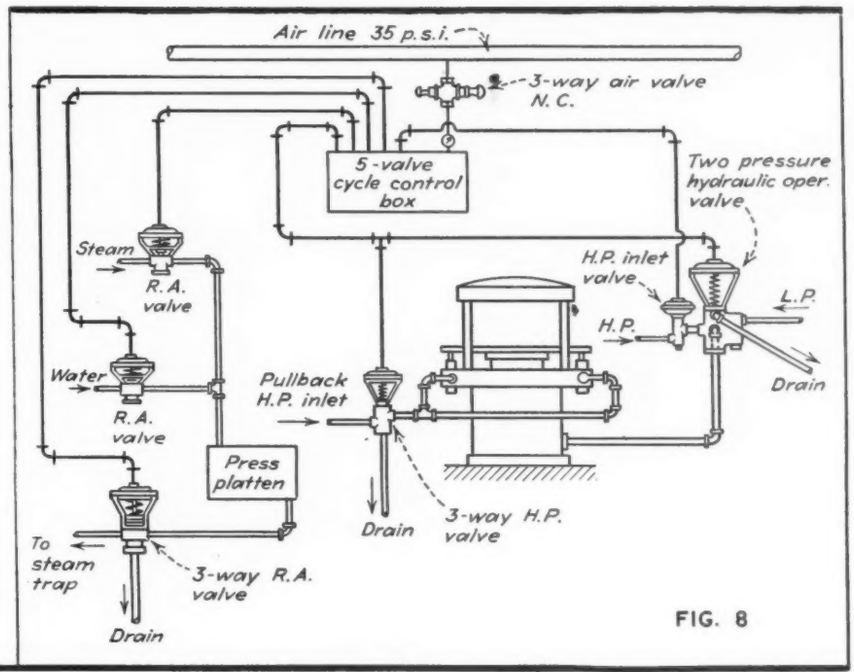


FIG. 8

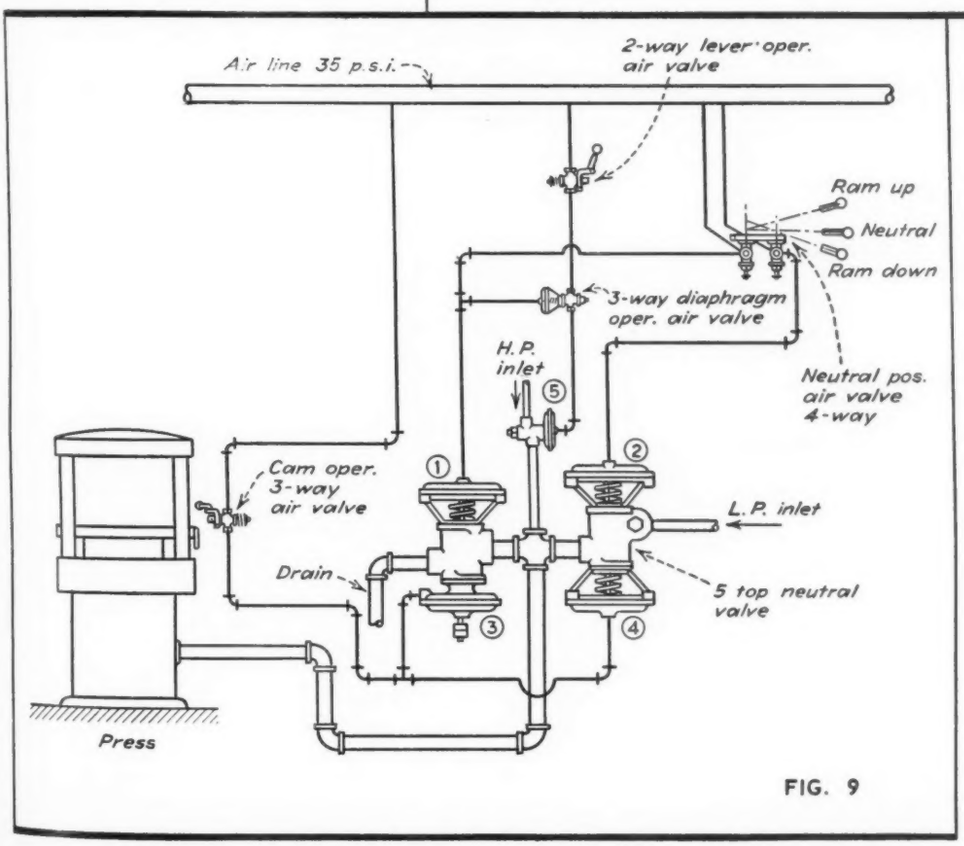


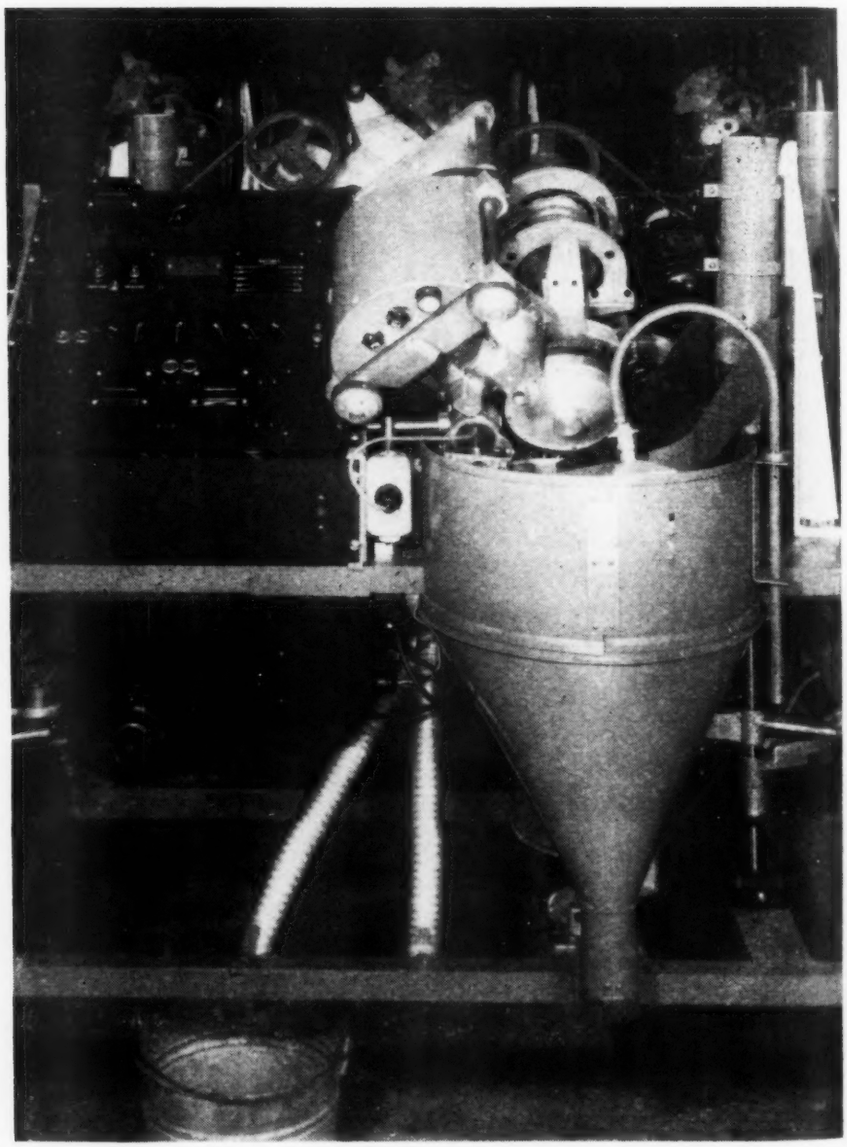
FIG. 9

Fig. 8 — Typical arrangement of hydraulic equipment and control valves for actuation by cycle controllers. Cycle controllers frequently are set completely to take over the operation of presses for curing many rubber products.

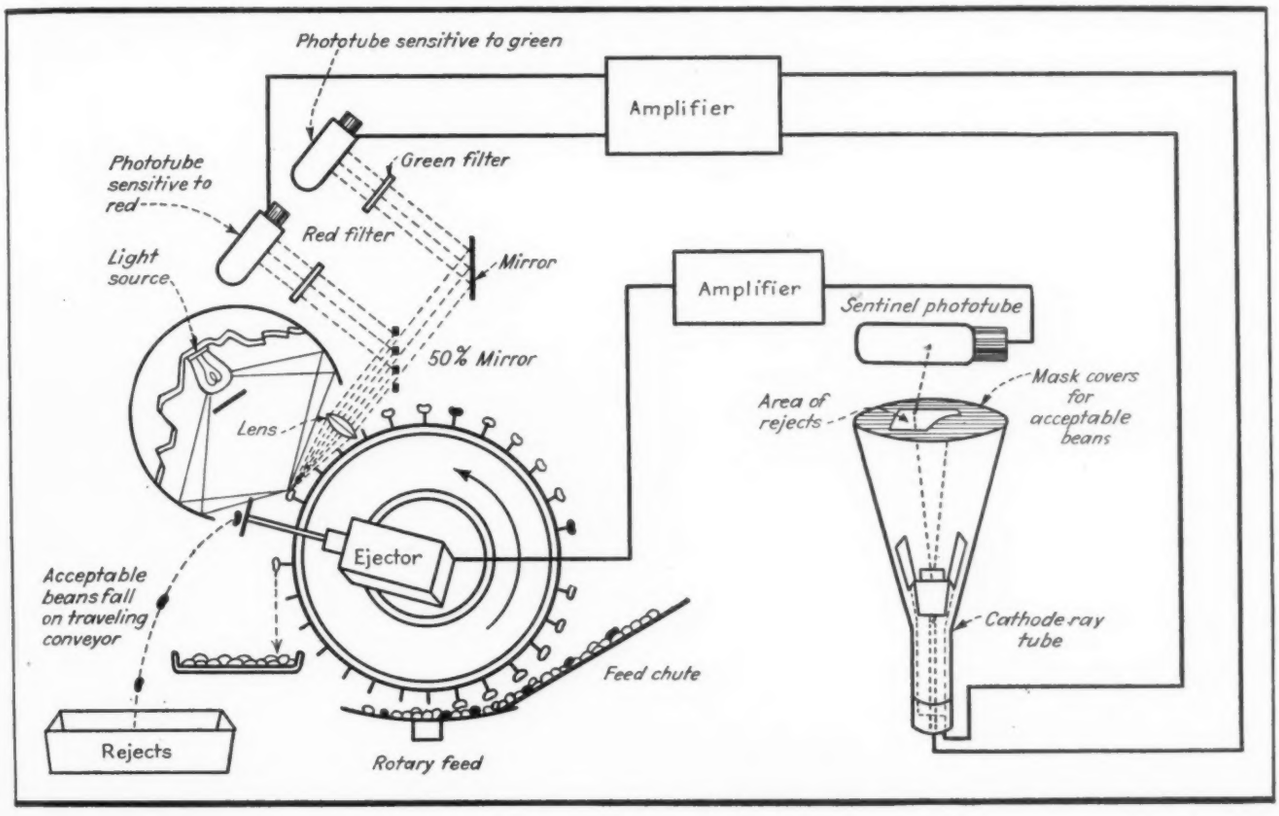
Fig. 9—For precision of press-ram movements specially designed control valves, such as this 5-diaphragm valve, are available.

PRODUCT DESIGNS

Rapid Color Sorting By Electronic Inspection



ELECTRONIC SORTING MACHINE, developed by the Electric Sorting-Machine Company, Grand Rapids, Mich., rapidly separates objects according to their color. The machine is used principally to examine foodstuffs such as peas, beans, lemons, peanuts, corn, seeds and coffee beans. Generally installed in batteries, each electronic sorter has a capacity of about 100 lb. of pea beans per hour, or 62 beans per second. Operation is continuous and automatic except for emptying the receptacles into which the discards are thrown. Two types of the machine have been designed: the "dark trip" and the "bichromatic." The dark trip type is sensitive to dark or light objects and can be adjusted to discard certain percentages of the objects inspected if they are in various stages of discoloration. The bichromatic machine is sensitive to all shades of a color or can be set to select one color or a number of colors from either end of the spectrum. White and various shades of yellow corn kernels can thus be sorted.

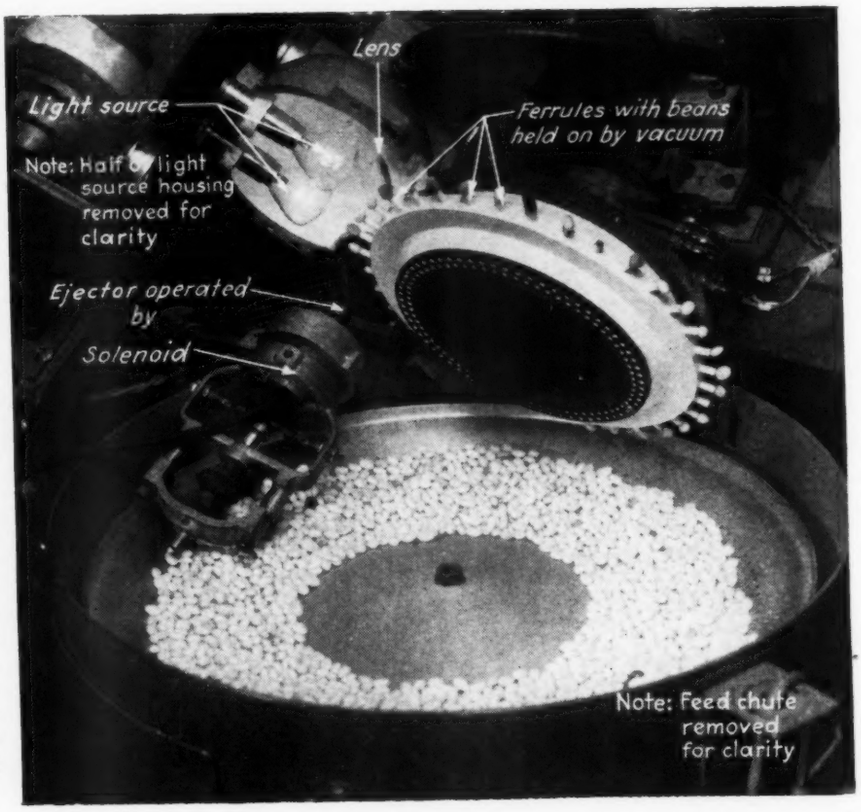


Acceptance or rejection of the object, say a bean, is based on the percentages of red and green light reflected by it. The machine feeds the beans one at a time before an optical system consisting of a light source, a focusing lens, two mirrors, two filters, and two phototubes. Light from the incandescent lamp falls on the bean and some of it is reflected through the focusing lens to the first mirror. This mirror reflects part of the light through a red color filter and onto a phototube sensitive to red light. Balance of the light hits the other mirror to be reflected through a green filter and onto a phototube sensitive to green. Output of each phototube is fed through dual channel amplifiers and from there to opposite deflection electrodes of a cathode ray tube. One phototube controls the vertical sweep of the cathode ray beam and the other the horizontal sweep. Degree to which the beam is deflected in either direction depends on the amount of the red or green light reaching the phototubes. A mask covers the "acceptable" area on the face of the cathode ray tube. Unacceptable beans deflect the electron beam off the mask so that the beam shines onto a sentinel phototube that operates an ejector mechanism to discard the faulty bean.

Battery of 32 machines examines 80,000 lb. in 24 hours. This means that 166,000,000 beans are viewed by photocell and accepted or rejected.

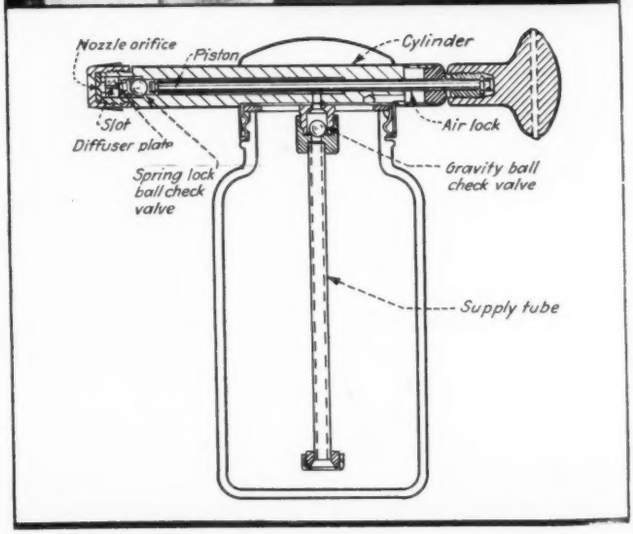
(Continued on next page)

Rapid Color Sorting (continued)


Lens
Light source
Ferrules with beans held on by vacuum
Note: Half of light source housing removed for clarity
Ejector operated by Solenoid
Note: Feed chute removed for clarity

Feed chute leads beans to a vacuum wheel having numerous tubes fixed on its periphery. A centrifugal blower causes a vacuum of 1 3/4 lb. to act through the tubes drawing a single bean onto each ferrule as it rotates at 107 r.p.m. through the feed chute. As the wheel turns it brings the bean before the light source. If not acceptable, the solenoid ejector, operating at sixty-three 5/8 in. strokes per second will push the bean into a "discard" receptacle. If acceptable, the vacuum is broken and the "good" bean drops onto a conveyor belt. Before the particular ferrule can pick up another bean, 150 lb. per sq. in. air pressure is blown through the tube to clean the orifice of any skin particles that might prevent reloading.

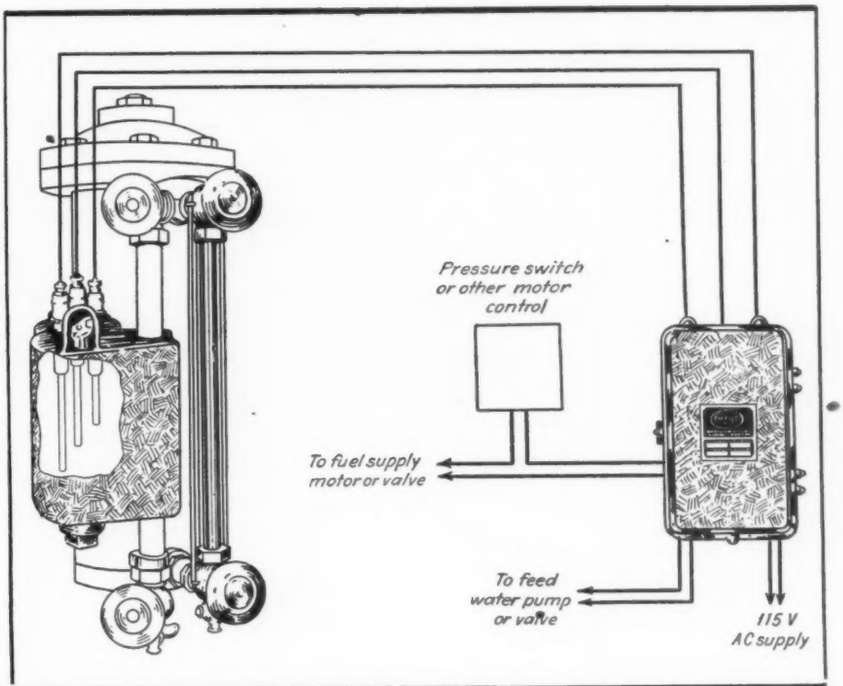
Compact Household Sprayer

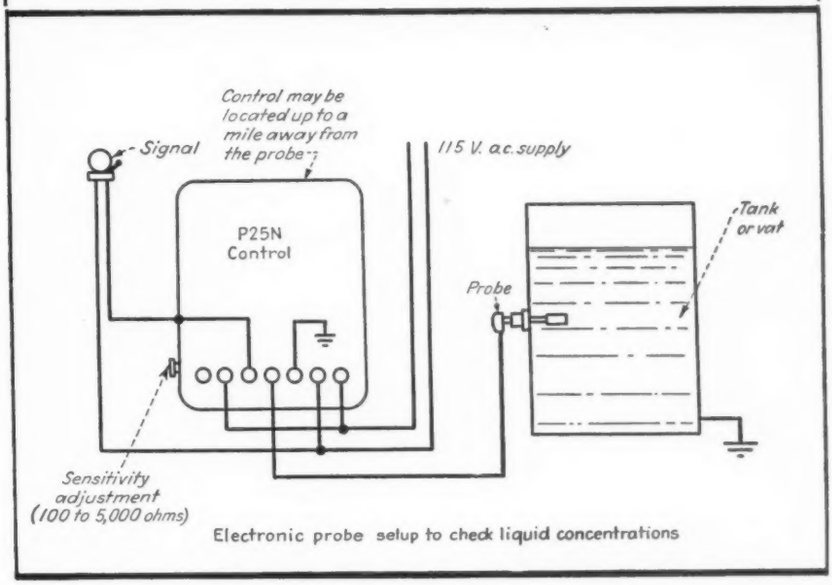


Ball check valves enable user to create a fog-like mist of antiseptics, deodorants, insecticides, or lotions at 300 lb. per sq. in. nozzle pressure with the manually operated household sprayer manufactured by the Cornelius Company of Minneapolis. Sprayer can be used at any angle without leakage or spillage. Corrosion is minimized since brass and stainless steel are used for the operating parts. Fluid is held in an 8 oz. glass container. Overall height is less than 5½ in., weight is 13 oz. No leather or rubber gaskets are used.

Operation begins when plunger is pulled back, and a springlock ball check valve closes off nozzle orifice thus preventing leakage at the nozzle. Vacuum caused by withdrawal of the plunger draws liquid past a gravity ball check valve to fill the supply tube. Forward movement of piston builds up pressure that closes the gravity ball check valve and opens the springlock ball check valve at nozzle. Liquid is vaporized as it is forced through four slotted grooves in a diffuser plate and thence through the 0.006-in. diameter nozzle orifice. Leakage around piston is prevented by an airlock. As liquid is drawn from the container, air enters the container through the airlock. This movement of air wipes the piston clean and carries excess fluids back to container.

Electronic Probe Controls Liquid Levels



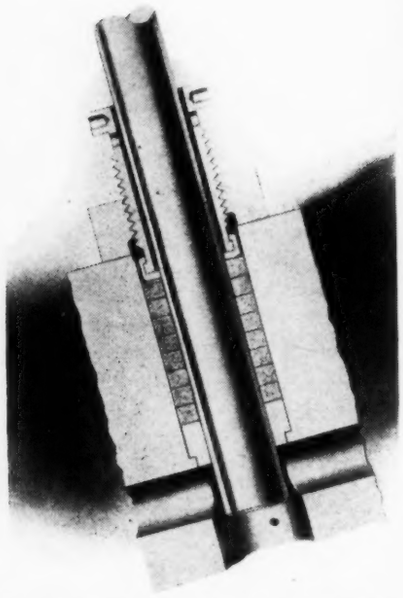

Control may be located up to a mile away from the probe
Signal
115 V. a.c. supply
P25N Control
Tank or vat
Probe
Sensitivity adjustment (100 to 5,000 ohms)

Electronic probe setup to check liquid concentrations

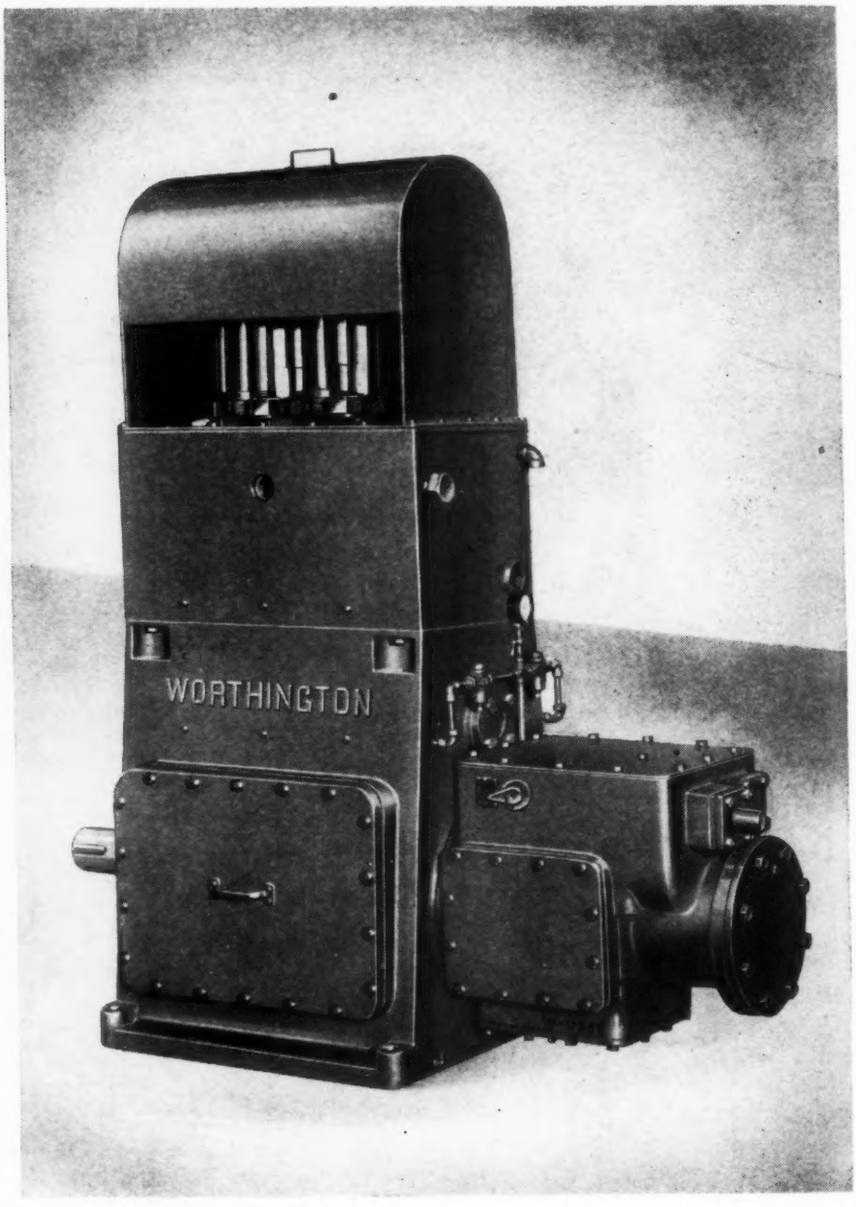
BOILER FEEDWATER LEVELS are regulated by the floatless Fireye Control that has no vacuum tubes and no moving parts. Built by the Combustion Control Corporation of Cambridge, Mass., the device has a metal probe extending into the tank so that water making contact with this probe relays to the control box a minute electrical current. Amplified, this current operates a heavy-duty switch to control valves, pumps or signals. Maintenance is insignificant because there are no floats, moving parts or stuffing boxes. Brass probes, mica insulated, are used for boiler feedwater control at pressures up to 300 lb. Voltage on the probe circuit never exceeds 25 volts, and so there are no shock or explosion hazards.

The Fireye boiler feeder control has three probes. Top and middle probes determine the high and low levels at which the feedwater pump is turned on and off, thus maintaining a constant water level. When water falls below the middle probe, the pump circuit is closed and water feeds into the boiler. When the level rises to the top probe, the water acts as a conductor of a minute electrical current at low voltage to operate the control and turn off the pump. If water falls below the lowest probe the system immediately shuts off the fuel supply and sounds an alarm.

Electronic level control is applied to other liquids with variations of this device built by Photoswitch Inc., an affiliated company. The Photoswitch electronic control checks liquid concentrations, maintains an interface between liquids and controls liquid levels. This control avoids the necessity of making frequent sample batch tests. Control adjustment is set so that the control relay will be actuated when the liquid or powder of a predetermined electrical conductivity contacts the probe. When the probe is immersed in liquids of other conductivities the control is inoperative; but when the exact desired conductivity is reached, the control functions to operate signals, valves or pumps. Accurate interface between two liquids can be maintained with this device by suspending the electronic probe to the desired level and adjusting the control to be sensitive to conductivity of the lower liquid, but insensitive to the conductivity of the upper liquid. Thus through operation of pumps or valves the interface can be held at this precise point. A change of 0.1 percent in the concentration of acids, bases, or salts can be detected. Two vacuum tubes are used. Stainless steel probes adapt this equipment to the sanitary requirements of the food processing industry.

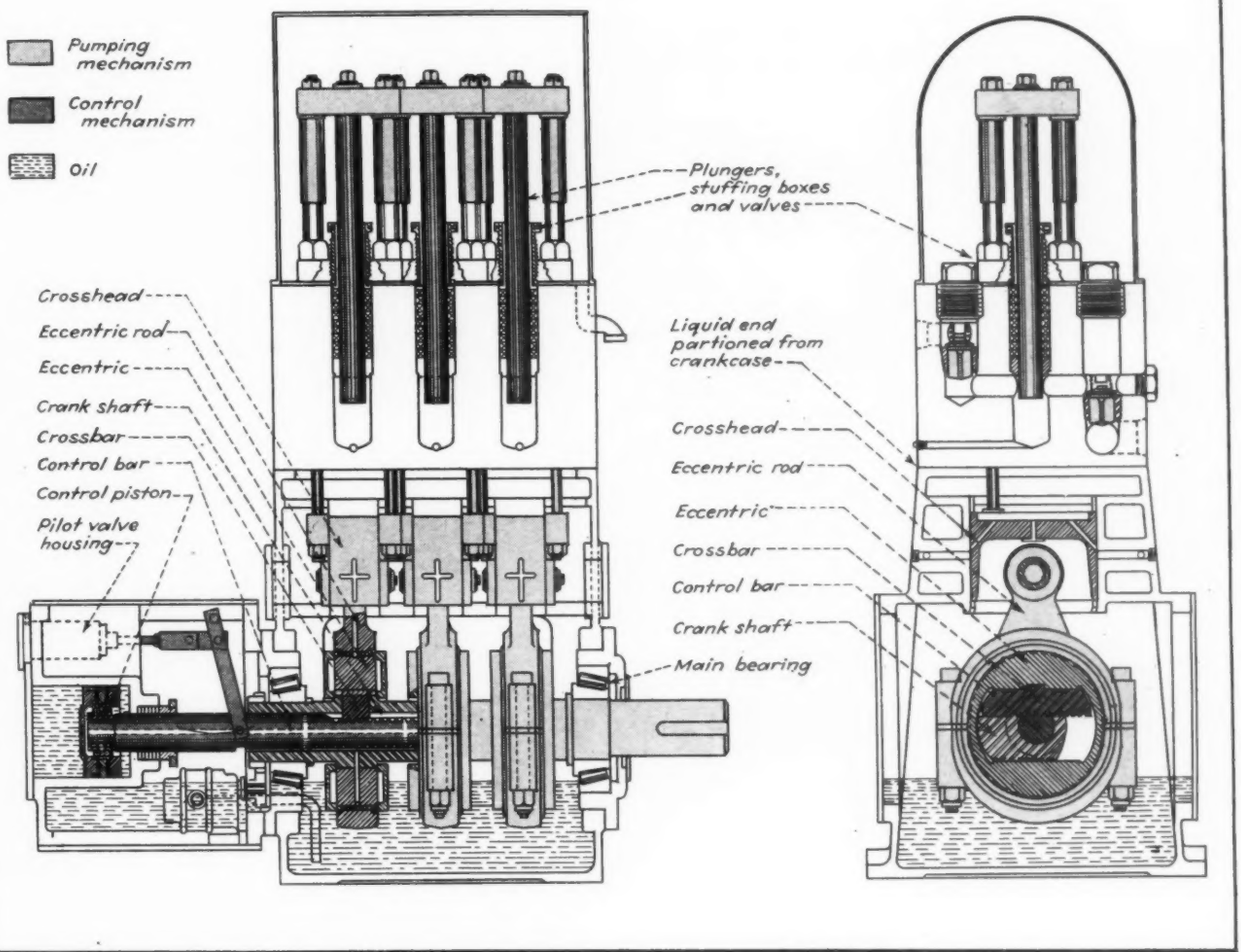


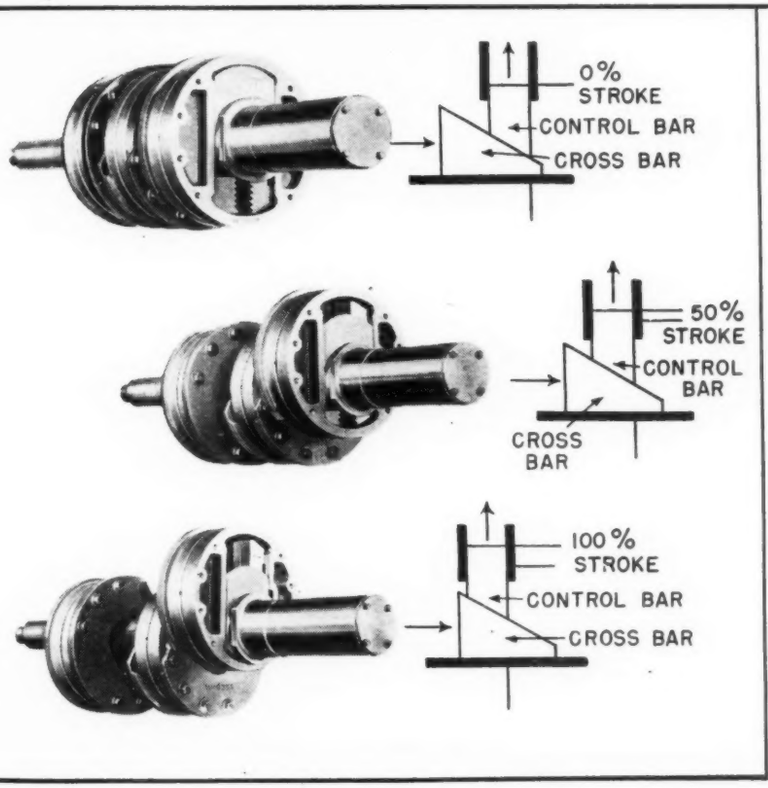
Flange and screw-type stuffing box gland is adjusted with a straight bar wrench without possibility of cocking the flange and thus reducing the life of the plungers and packing.



Eccentric Position Controls Full Range of Pump Output

STEPLESS, zero to 100 percent capacity flow in a high-pressure pump has been achieved in the Automatic Variflow Triplex pump produced by the Worthington Pump & Machinery Corporation, Harrison, N. J. The pump has been redesigned to use the same frame as the corresponding size of conventional pumps. Ease of maintenance has been aided by keeping the stuffing boxes on top of the cylinders where they can be packed and adjusted without difficulty. Outboard location of the stuffing boxes precludes the possibility of normal stuffing box leakage finding its way into the crankcase of the pump. The crankshaft is in the base of the pump in a fully-inclosed frame. Bearings are of the tapered roller type with positive automatic pressure lubrication provided by a direct-driven gear pump. The stroke changing element and bearings are flooded with filtered oil at all times. The oil pump also provides pressure for operating the hydraulic stroke control cylinder. The crankshaft is made of forged steel, the alloy steel control bar is hardened. The crossbar is phosphor bronze, and the eccentric rod is made of annealed cast steel. The pump is designed to be direct driven or belt driven. It is suitable for all free-flowing liquids. A discharge pressure of close to 10,000 lb. per sq. in. is claimed. A constant speed motor or turbine may be used to drive the pump, regardless of the capacity requirements, because pump output is variable from 0 to 100 percent by means of a simple, compact mechanism.

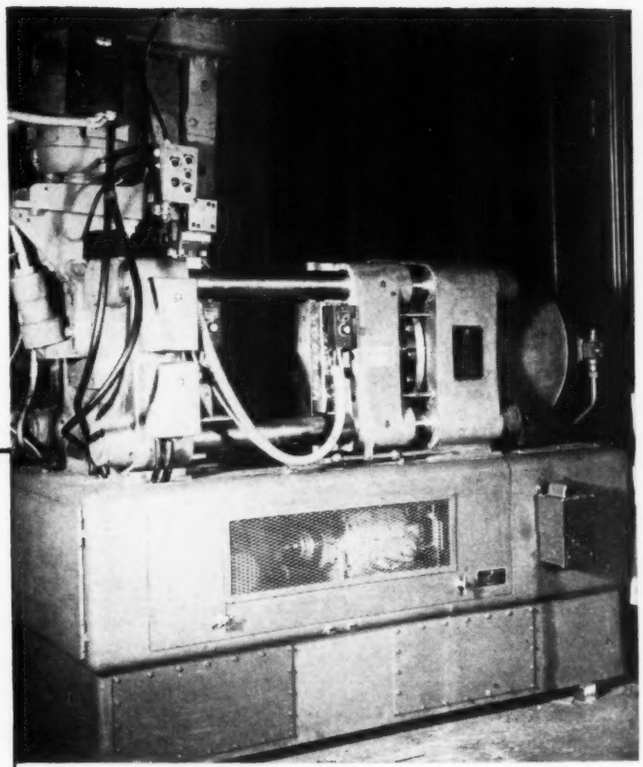


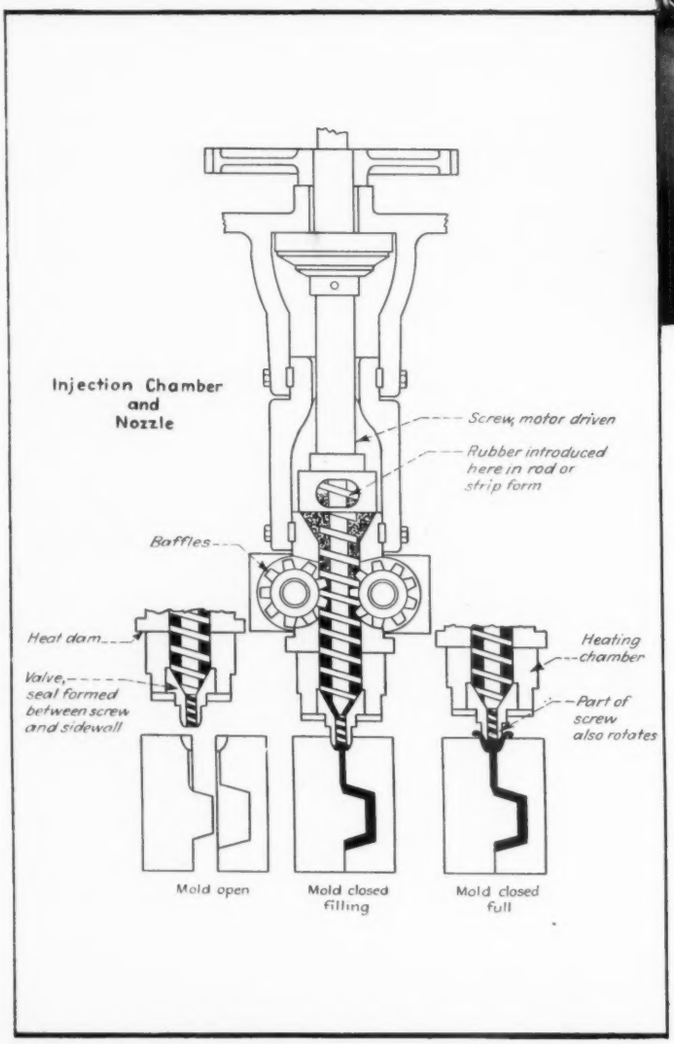


Stepless variation of the pump's output is achieved by changing the stroke of the plunger. This is based on the principle of shifting the position of an eccentric to suit the desired stroke. Positioning the eccentric is done by an hydraulically motivated control bar sliding in the hollow crankshaft. The control bar engages the crossbar that positions the eccentric. Oblique-tooth racks, operating as a series of inclined planes, form the engagement between the control bar and crossbar. As the hydraulic control slides the control bar toward the pump, the crossbar and eccentric are forced radially outward thereby increasing the eccentricity. This has the effect of lengthening the eccentric rod so that the crosshead is driven through a greater stroke. Thus the stroke of the pump is adjusted. Moving the control bar away from the pump produces the opposite results until ultimately the crossbar centers the eccentric on the axis of the crankshaft. At this point there is no stroke and the output of the pump is zero.

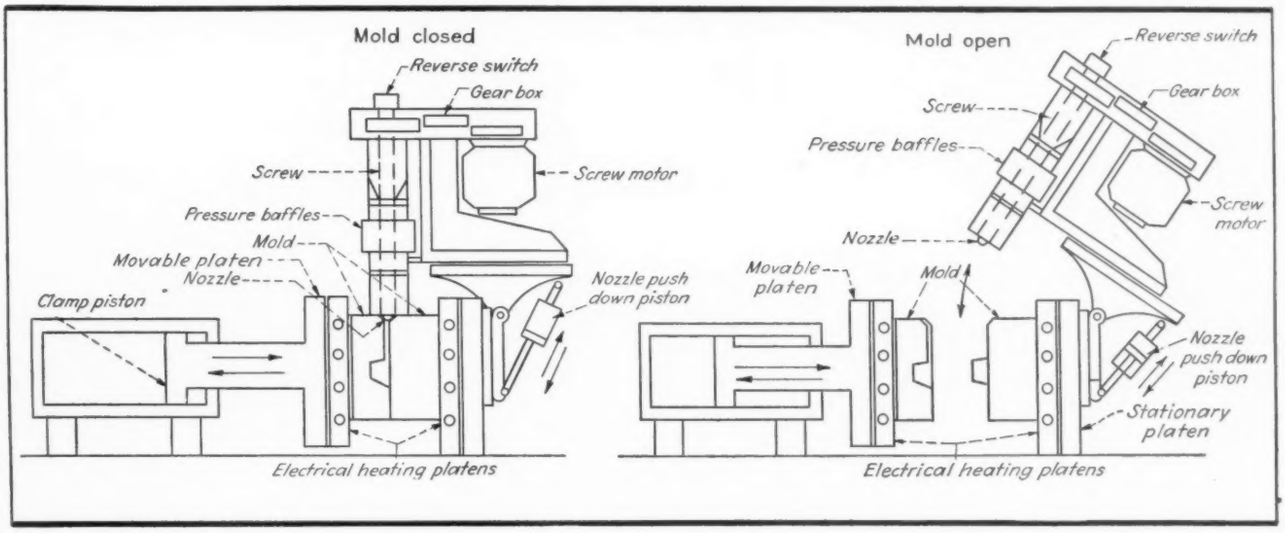
Extrusion Principle Applied to the Injection Molding of Rubber

NATURAL OR SYNTHETIC RUBBER can be injection molded with the HPM Turbojector developed by the Hydraulic Press Manufacturing Company, Mount Gilead, O. Rubber parts so molded are claimed to have longer life and greater tensile strength than those molded by conventional compression methods. Preforms are not required. Curing time is reduced as much as 93 percent. Finishing operations are minimized because there is no flash to remove. The machine is hydraulically controlled and fully automatic. The head of the HPM Turbojector consists of three sections: driving mechanism, injection chamber and nozzle. Driving mechanism is a ten hp., four-speed, high-torque electric motor with a gear transmission to rotate the extruding screw. Four speeds are available because it is desirable to vary the screw feed rate for different types of rubber. Ninety-six r.p.m. is the most generally used speed.





Rubber in any solid form is fed into the injection chamber by the feed screw. Near the middle of the screw its profile changes to one that resembles an Acme thread. At this point are free-running baffle gears that act as pressure locks. They prevent the rubber from turning with the screw and from backing up into the feed section. When passing through the injection chamber the rubber may develop temperatures as high as 300 deg. F. as a result of the frictional head created by the screw and baffle gears. Additional heat is secured at the nozzle from an electric resistance band heater. Heating the rubber in the injection chamber and nozzle accounts for the rapid curing cycles that are obtained. The nozzle contains an automatic check valve to prevent the rubber from backing up into the screw chamber after the mold is filled. Injection pressures can run as high as 18,000 lb. per sq. in. on the material to be molded. Maximum injection capacity is eight pounds. As the mold is filled, rubber extruded from the nozzle tilts the injection unit slightly, causing an electrical contact that momentarily reverses the screw and stops its rotation. The hydraulic ram tilts the injection unit and reseats the nozzle. This closes check valve in nozzle, preventing the rubber in the mold from backing up through the sprue bushing.



Hydraulic mold clamp, consisting of a double-acting ram fitted with metallic piston rings working in a smooth bored steel cylinder, controls the moving clamp platen. Main ram is equipped with a small internal booster ram that closes the clamp platen to within a fractional part of an inch of the total clamp stroke. At this predetermined position, hydraulic fluid under pressure is shifted to the large ram area. This arrangement permits practically all of the clamp travel to be accomplished by the use of the fast-closing booster ram assuring automatic slowing.

Random Notes

Duraluminum suitcases of sheet metal taken from fallen airplanes are being made in France. The outside of the suitcase is crinkled in an imitation of leather. The inside is given a light coat of glue and then sprayed with wool dust, a process of flaking already known from plane making.

* * *

If the 6-cylinder opposed piston liquid-cooled Jahco motor performs as well under road testing as it did on the block, the engine will power the light-weight Ford car. The Jack and Heintz Precision Industries, Inc., Cleveland, motor gets 95 hp. from its 126 cu. in. displacement and operates up to 6,500 rpm. Fuel consumption is claimed to be 25 percent less than the conventional motor of comparable size. The block is of die cast aluminum.

* * *

A 30,000 ton forging press for magnesium forging was built at the I. G. Farben plant in Germany during the war. The press was equipped with 8 supporting columns and 8 cylinders were used to distribute hydraulic pressure evenly. Press stood 85 feet above the floor line, dies were mounted on a table about 33 by 20 feet. At the time production ceased, the press was forging wing span caps 18 to 30 feet long for aircraft. Other remarkable forgings were a large landing wheel and a side frame for 155-mm. gun. In addition to extrusion forgings and rolling processes known in the United States, the Germans developed a water process for making ingots. The process embodied a hot thin wall mold. The water dip method was claimed to have the advantages of better uniformity of composition, less waste, consistent quality, less dependence on skilled labor, and no limit on the ingot size. Principle disadvantage was higher cost.

* * *

Less than 10 percent of the torque applied to tighten a nut is reflected in increased tension in the bolts. Therefore, large diameter long bolts are sometimes made hollow, an electric resistance unit is inserted and the nut is screwed tight while the bolt is hot. Shrinkage develops the desired tension.

* * *

Custom-styled convertible coupes are to be mass produced under the guidance of Howard Darrin, West Coast designer of custom-made cars. The convertible is powered by a 6-cylinder Continental engine; body consists of four stampings. Hydraulically operated windows and torsion bar suspension are among the other features. The Darrin weighs 2,400 pounds.

* * *

A Chinese typewriter has been developed by the International Business Machine Corporation, Poughkeepsie, N. Y. The machine will write 5,400 characters and can type horizontal lines from left to right, the new Chinese method, or type vertical columns from right to left in the ancient tradition. The 5400 characters are stamped on a revolving cylinder in a machine a little more than twice the size of a standard typewriter. The character is selected and printed electrically through two side-by-side keyboards with two banks of keys each. The keys are numbered by banks and 4 keys must be hit to print each character. An experienced operator can type 45 words a minute.

* * *

Steam ship funnels are slated to disappear in the streamlined postwar passenger ships according to George C. Sharp, one of the nation's foremost ship designers. Since petroleum became the principal fuel for large ships, space consuming smoke stacks have

become an achronism that persisted only because of tradition. Tall slender smoke stacks disguised as king posts are ample to dispose of engine-room soot but they must be kept high in order to allow the wind to carry off the soot before it settles on the top decks.

* * *

Automobile production for 1947 is estimated by the most optimistic automobile executives to be a total of 4,800,000 cars and trucks. Even though plant facilities could handle 6,000,000 units, the sheet metal supply will not cover the higher figure. Other steel shortages currently holding up production are spring wire, bolts and nuts. At least one manufacturer will offer independently sprung rear wheels on his 1947 models.

* * *

Catapulting the pilot from high speed planes is the latest safety device developed by the Army Air Forces, Wright Field, Ohio. The pilot and seat are thrown 50 ft. above the plane and the parachute opens automatically.

* * *

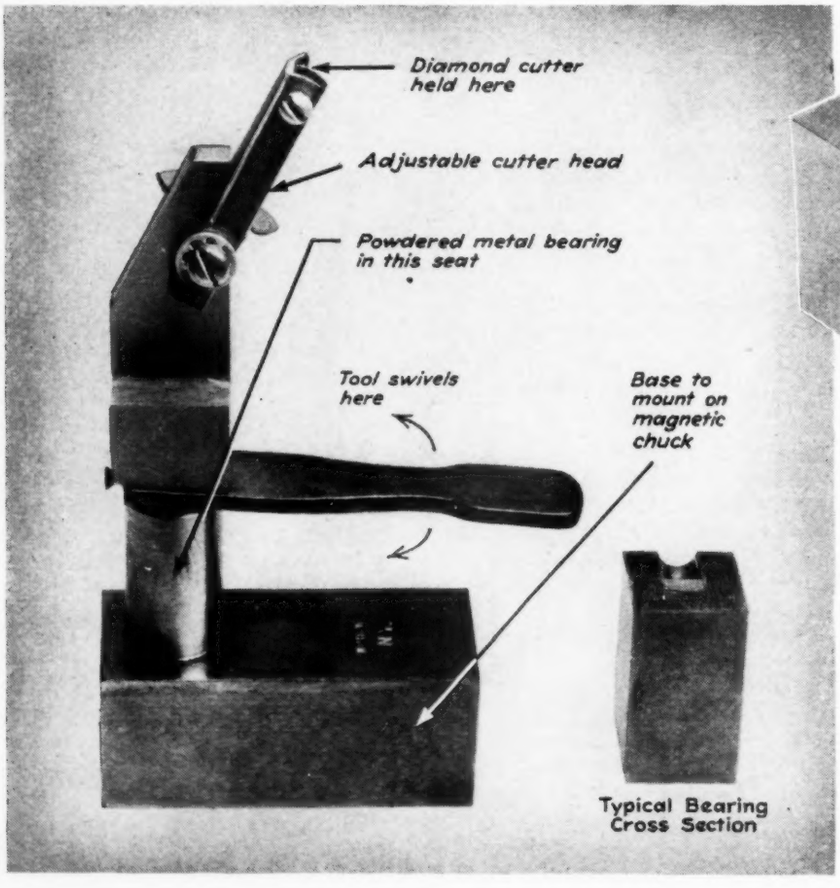
A protective coating especially effective against salt water corrosion has recently been applied to the Golden Gate Bridge as a substitute for paint. Powdered zinc is mixed with gas and by means of a gun, forced through a flame of 785 deg. F., thus becoming fused with the cleaned and blasted steel surface. A layer of zinc 0.010 inches thick is deposited.

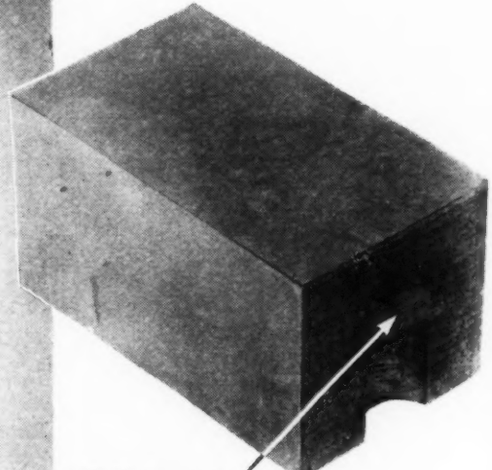
* * *

A phosphorescent compound, Lustron 1200, has been developed by the Monsanto Chemical Company. It is a phosphorescent molding material that has an afterglow six to eight hours after exposure to ultra violet light. The material is expected to be used on instrument panels, door numbers, and markers.

Powdered Metal Cold Formed into Bearings

COLD PRESSED powdered metal bearings, formed without sintering, are made by the National Diamond Hone and Wheel Company of New York. Pressing is done by a special technique using a series of sharp blows to approximate a pressure of eight tons per sq. in. The powder forms an homogeneous mass of mechanically bonded particles with complete cohesion. Compressive strength is good, tensile strength is low. The bearings are used in a diamond radius wheel dresser to make a cheap and simple bearing seat for the dresser head. Oil impregnation gives an hydraulic cushioning that prevents chatter marks.


Diamond cutter held here
Adjustable cutter head
Powdered metal bearing in this seat
Tool swivels here
Base to mount on magnetic chuck
Typical Bearing Cross Section

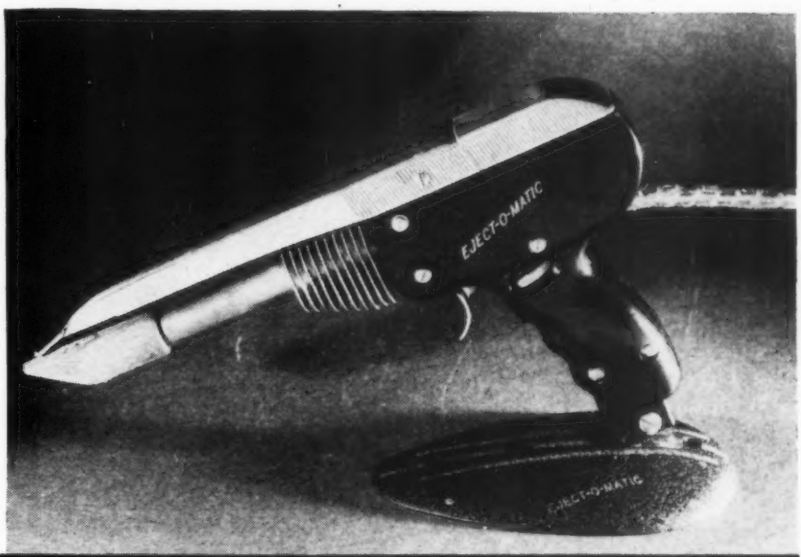


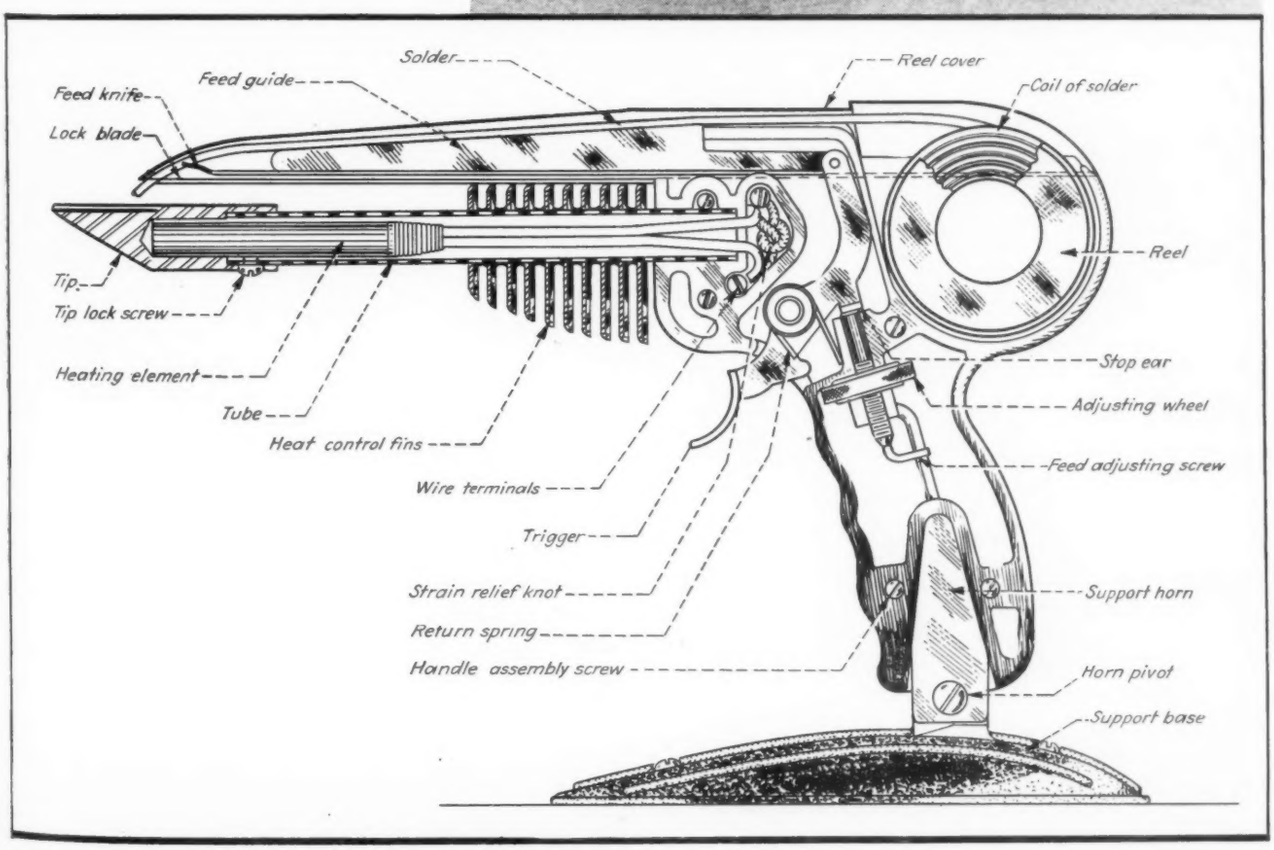
Bearings have been successfully made from copper-nickel-lead alloys, copper-tin, and pure copper. It is believed that the sharp metallic blows allow surface oxides to be ruptured thus enabling an intimacy of contact that promotes atomic welding. The bearings so formed can be impregnated with oil.

Trigger Feed Soldering Iron

AN AUTOMATIC feed soldering iron for one hand operation in home or factory has been designed by the Hyde Engineering Company. The iron houses a solder reel and feeds a preselected amount from 1/16 in. to 3/8 in. of solder to the tip each time the trigger is pulled. Solder from 1/16 in. to 3/32 in. d. can be used. The safety utility base has a horn that supports the iron at any angle. The adjustable base combined with a foot operating device and 5 lb. reel of solder (not shown) adapts the iron to production line use where the operator must have both hands free. The Eject-O-Matic soldering iron is manufactured by the Multi-Products Tool Company of Newark, N. J. Stainless steel is used for most parts other than the plastic handle.

Heating element is in the tip so that heat flows to the work with half the wattage required by the usual installation in the handle. The tip has successive coatings of non-corrosive alloy that does not erode with the solder. Brass heat-control fins discard the amount of heat necessary to keep tip temperature under 750 deg. F. Thus the flux will not be carburized rapidly. Simple trigger mechanism ejects the solder. Upon release of the trigger, the mechanism withdraws the solder supply so that an excess amount will not melt by conduction. Rotation of the adjusting wheel changes the amount of angular movement of the trigger and hence the amount of linear movement of the feed knife.





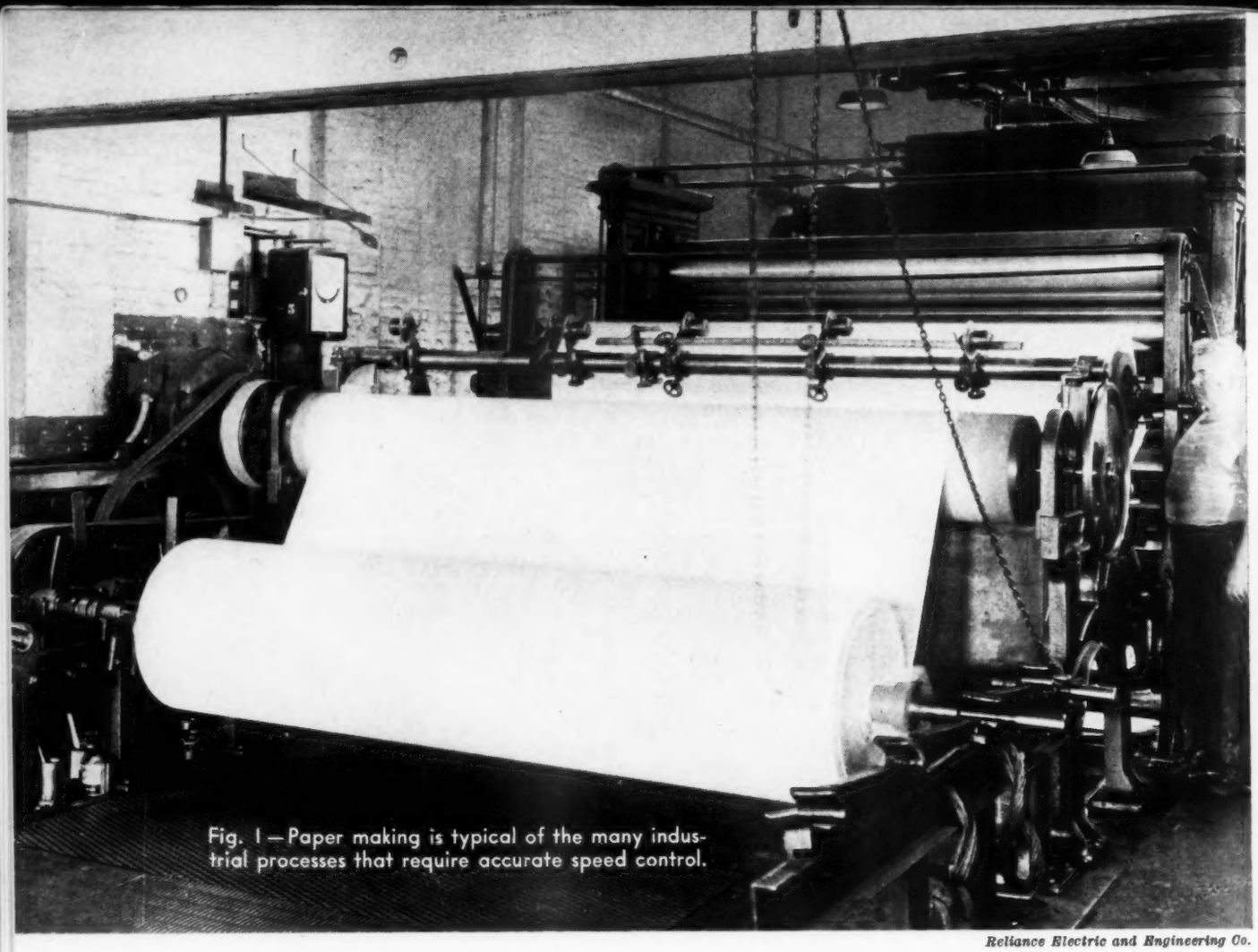


Fig. 1—Paper making is typical of the many industrial processes that require accurate speed control.

Reliance Electric and Engineering Co.

Constant Speed Regulation

Four Systems for Maintaining Preset Speed

J. E. JONES
Supervising Engineer, Cutler-Hammer, Inc.

Fundamentals of four types of constant speed regulating systems commonly used in industry are explained and discussed. The general principles of operation and relative field of application are given for the polarized-relay, solenoid operated, electronic, and rotating systems.

MANY PROCESSES in the paper, food, chemical, textile and other industries require that the speed of the machine when once set at a certain value continue to run at that speed with practically no variation. The machines for such processes, as shown in Fig. 1, are usually driven by d.c. motors with a wide range of operating speed that are supplied from an adjustable-voltage d.c. generator. In addition to the usual adjustable-voltage system for starting, inching, and speed setting, additional control is used for maintaining the preset speed while running.

The preset speed is maintained within close limits automatically. In the adjustable-voltage system the speed can be automatically regulated by controlling either the motor or generator shunt field. In the constant voltage d.c. system, the shunt field of the driving motor is controlled by the automatic regulator.

The operating speed is usually preset by a field rheostat or a presetting rheostat. Adjustable-voltage systems frequently use motor-operated rheostats and the desired operating speed is obtained by pressing the conveniently located "Increase" or "Decrease" pushbuttons. Other control schemes use small potentiometer-connected rheo-

stats as speed presetters. Regardless of the speed adjusting or presetting method used, the function of the automatic speed regulator is to maintain the desired speed, despite fluctuations in the load, heating of the machines, or variations in the armature and field voltages.

All automatic systems require some control impulse, which is usually electrical, to serve as a measured value that is proportional to the thing being regulated. Speed control systems usually make use of small d.c. generators, called tachometer generators or pilot generators. These small generators, which have a straight-line relationship between speed and generated voltage, are driven by the machines that are to be regulated at constant speed. The output voltage of the pilot generator is a direct measure of speed and can be used as the measured quantity to actuate the automatic speed regulator. The automatic regulator acts to maintain the pilot generator voltage constant at some value, and in so doing it holds the machine speed constant.

Polarized-Relay Scheme

When extreme accuracy is not required, the polarized-relay control is perhaps the simplest that can be used. A sensitive polarized relay detects variations from the preset speed and actuates contacts that restore the proper speed balance. The accuracy of performance depends upon the sensitivity of the polarized relay used. The polarized relays usually have a sensitivity of plus or minus one volt and are suitable for circuits up to about 200 volts. This provides rather accurate control at high speeds and correspondingly reduced accuracy at lower pilot generator voltages. For a 5 to 1 speed range, the pilot generator voltage might vary from 200 down to 40 volts, depending upon the preset speed. Accuracy of the system also depends upon the number of speed points on the corrective rheostat that responds to the polarized relay impulses.

The simplified connections for a typical polarized relay constant-speed circuit are shown in Fig. 2, in which all contactors and relays that are not essential to the constant-speed feature are purposely omitted. The d.c. motor gets its power from an adjustable-voltage generator. A constant voltage exciter furnishes d.c. for the motor and generator shunt fields and to operate the control devices. A pilot generator is coupled to the motor and its voltage output is used as the necessary control impulse.

The voltage of the main generator is controlled by a potentiometer-connected generator field rheostat. This field rheostat is driven by a small pilot motor that is provided with forward and reverse field windings. These field windings are connected to the single-pole double-throw contacts of the polarized, *PR*, relay. Therefore, the *PR* relay can operate the generator rheostat in either direction to increase or decrease the main generator voltage as required.

The speed setter is a potentiometer rheostat connected across the excitation bus. The movable contact is connected in series with the two actuating coils of the *PR* relay and thence to one terminal of the pilot generator. The pilot generator opposes the speed setter voltage. When the system is in balance these voltages are equal and no current flows in the *PR* relay coils.

With an increase in load, the main motor tends to slow down. The pilot generator voltage decreases a tiny amount and current flows through the *PR* relay coils. This causes the *PR* relay to make a circuit through its increase contact, thereby operating the generator-field rheostat in the proper direction to increase the generator voltage. This results in a speed increase and restores the system to its balanced condition. This action is entirely automatic and within the limits of the accuracy of the devices used, holds the speed of the driving motor constant.

Solenoid-Operated Regulator

Solenoid-operated regulators are commonly used to regulate the voltage of generators and alternators. Although they are usually referred to as rocking-contact regulators, carbon-pile regulators, or the like, they are actually solenoid-operated field rheostats. The solenoid is connected to the voltage that is to be regulated and its plunger works against a calibrated spring. Movement of the plunger is transmitted to the regulating resistor through some type of sensitive mechanism. Small variations in the regulated voltage cause the plunger to move slightly. These plunger movements change the regulating resistor in such a way as to maintain the desired voltage.

The basic circuit commonly used for constant-speed regulation of a single motor on drives, such as a single motor paper making machine, is shown in Fig. 3. The motor is large enough to drive the entire machine and the various sections are clutched-in to a line shaft that runs the entire length of the machine. This simplified diagram shows only those circuits that are essential in the speed regulating circuit. A rocking-contact type speed regulator is shown simply because its operation is easily pictured schematically.

As shown, the main motor receives power from an adjustable-voltage generator. In addition, there is a pilot generator driven from the main motor, or the machine. This pilot generator has a straight line relationship between speed and generated voltage. Therefore the machine speed is maintained constant by maintaining the voltage of the pilot generator constant.

The pilot generator armature circuit is connected to the motive coil of the rocking-contact regulator through suitable resistors. A balance is obtained at 0.75 amp. through the solenoid coil of the rocking-contact regulator. This means that the resistance of the pilot generator armature circuit must be correctly proportioned so that at all speeds, this value of 0.75 amp. will flow in the circuit.

The speed-setting rheostat consists of 3 rheostat sections operated in tandem as shown in Fig. 3. The pilot generator sec-

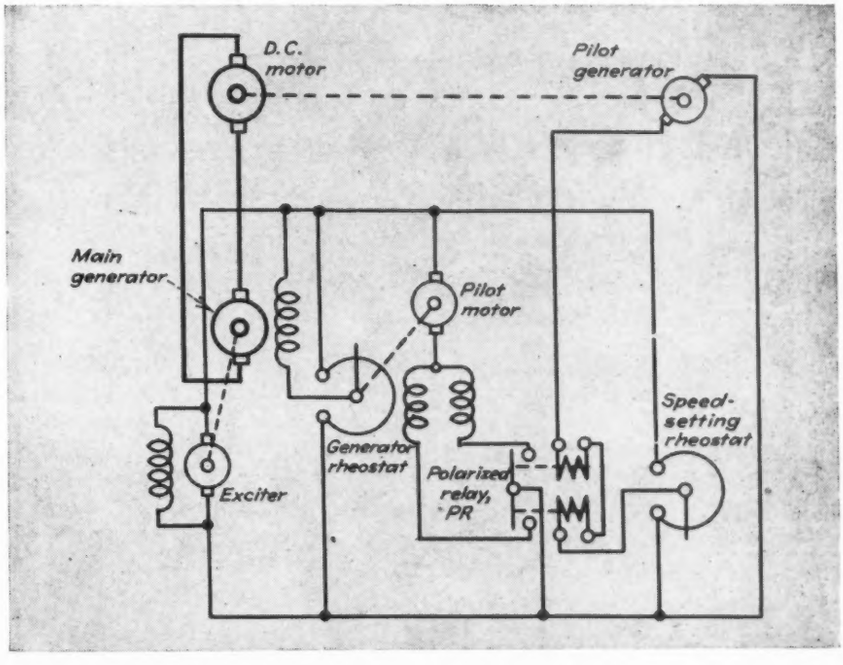


Fig. 2—Simplified diagram for controlling speed by a polarized relay.

tion has resistance over its entire travel, and is designed to maintain 0.75 amp. in the pilot generator circuit at all speed settings. The generator field rheostat section is shown as having speed control from the lowest speed position over approximately two-thirds of its travel. This represents the entire speed range by generator field control.

The motor field rheostat section holds the motor field at a constant strong field value over the adjustable-voltage portion of the speed range. It has resistance for weakening the motor shunt field for additional speed increase over the last one-third of travel. These three rheostat sections must be properly proportioned so the speeds throughout the range, as established by the generator and motor rheostats, agree approximately with the speeds called for by the pilot generator section.

The speed regulator acts as a voltage regulator and operates to maintain the voltage across its motive coil at a constant value. For example, with the balanced condition in Fig. 3, assume the load on the motor increases slightly tending to slow down the drive. This tendency to slow down will result in a very small reduction in the pilot-generator voltage that may change the circulating pilot-generator current from 0.75 to 0.745 amp. This unbalances the regulator causing its rocking contact to move slightly in the direction to weaken the motor shunt field. This, restores the regulator to its balanced condition and the speed to the desired value. Good regulators maintain a balancing voltage with an accuracy of plus or minus one-half percent.

This type of speed regulation can be applied to any machine, and the regulator can be used directly in the field circuit for motors within the wattage capacity of the regulator. For larger motors, exceeding perhaps 200 hp. and depending upon the field characteristics it may be necessary to add an auxiliary exciter to operate in conjunction with the regulator. The regulator itself controls about 20 percent motor shunt field range when used as shown in Fig. 3. The solenoid-operated regulators require relatively large pilot generators since, in some applications, the regulator coil current may be as much as one ampere.

Electronic and Rotating Regulators

A number of systems using electronic devices are in use today. No attempt will be made to go into the detailed circuits of any of these systems. However as shown in Fig. 4, most of them, when reduced to their basic elements, operate on about the same principle. Here again, the speed of a d.c. motor is to be held constant at some value. The motor armature is connected to a suitable d.c. supply and its shunt-field current is supplied electronically.

The pilot generator that is driven by the main motor can be a small device since the control current required from it is minute. The terminals of the pilot generator are connected to a potentiometer rheostat and the sliding contact of this rheostat is connected to the electronic control.

It is necessary to have a reference voltage with which the pilot-generator voltage is compared in order to maintain constant speed. This reference voltage must be carefully regulated since it serves as a standard. It may be supplied from a master pilot generator but more frequently is supplied electronically.

When this circuit is in operation, the pilot-generator voltage is balanced against the reference voltage. Any differences in these voltages are detected by sensitive electronic tubes. The result is amplified and the main motor shunt field current is modified electronically to restore a balance of the voltages. The speed of the main motor can be changed by changing the setting of the small potentiometer rheostat since this alters the portion of the pilot generator voltage that is compared with the reference voltage.

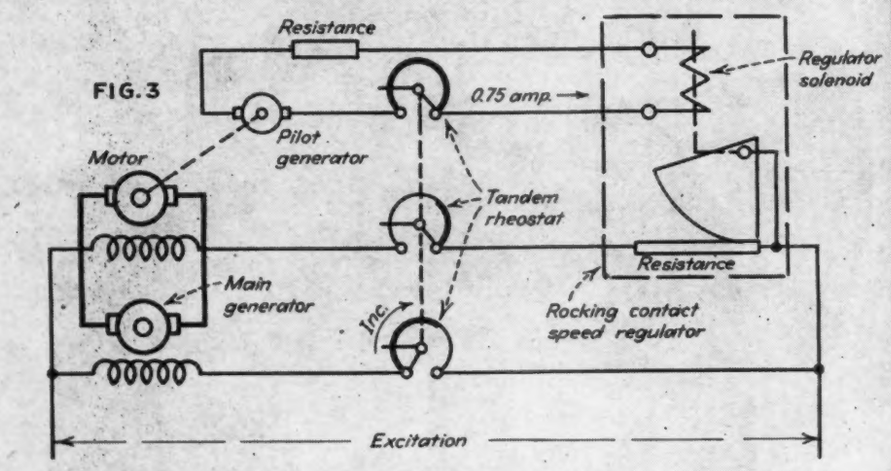
In recent years the rotating type of regulator has come into rather general use. Several types, having detail differences but generally alike in principle, are being applied by designers. Essentially, they are small d.c. generators that are usually driven at constant speed and have two or more shunt field windings. They serve as efficient power amplifiers, since small impulses of electrical energy when fed into the shunt field windings, control the watts output of the armature up to its rating. In general, rotating regulators are used on applications where the driving motors are large and have shunt fields requiring regulator capacities larger than are provided by other types of regulators.

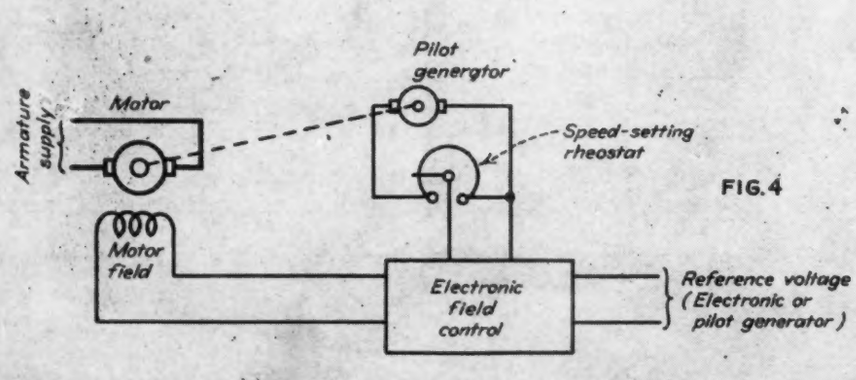
Rotating regulators, like the other types, are voltage regulators and operate in the speed-regulating circuits from pilot generators. They can be used in any of the schemes discussed for other regulators with detail circuit variations depending upon the design of regulator being used. Sometimes, the electronic scheme is combined with a rotating regulator, using the electronic method for sensitive detection of speed variations. The output of the electronic regulator is then fed into a field of the rotating regulator to obtain additional power amplification. The rotating regulator armature then controls the shunt field of the driving motor in such a manner as to maintain constant speed.

The polarized-relay, solenoid-operated regulator, electronic and rotating regulator are the most commonly used types in industry today. A clear understanding of the basic principles may often be useful in helping designers select and apply the type best suited for each new speed control problem.

Fig. 3—Constant speed control by a solenoid operated regulator.

Fig. 4—Schematic diagram for speed control by voltage comparison.





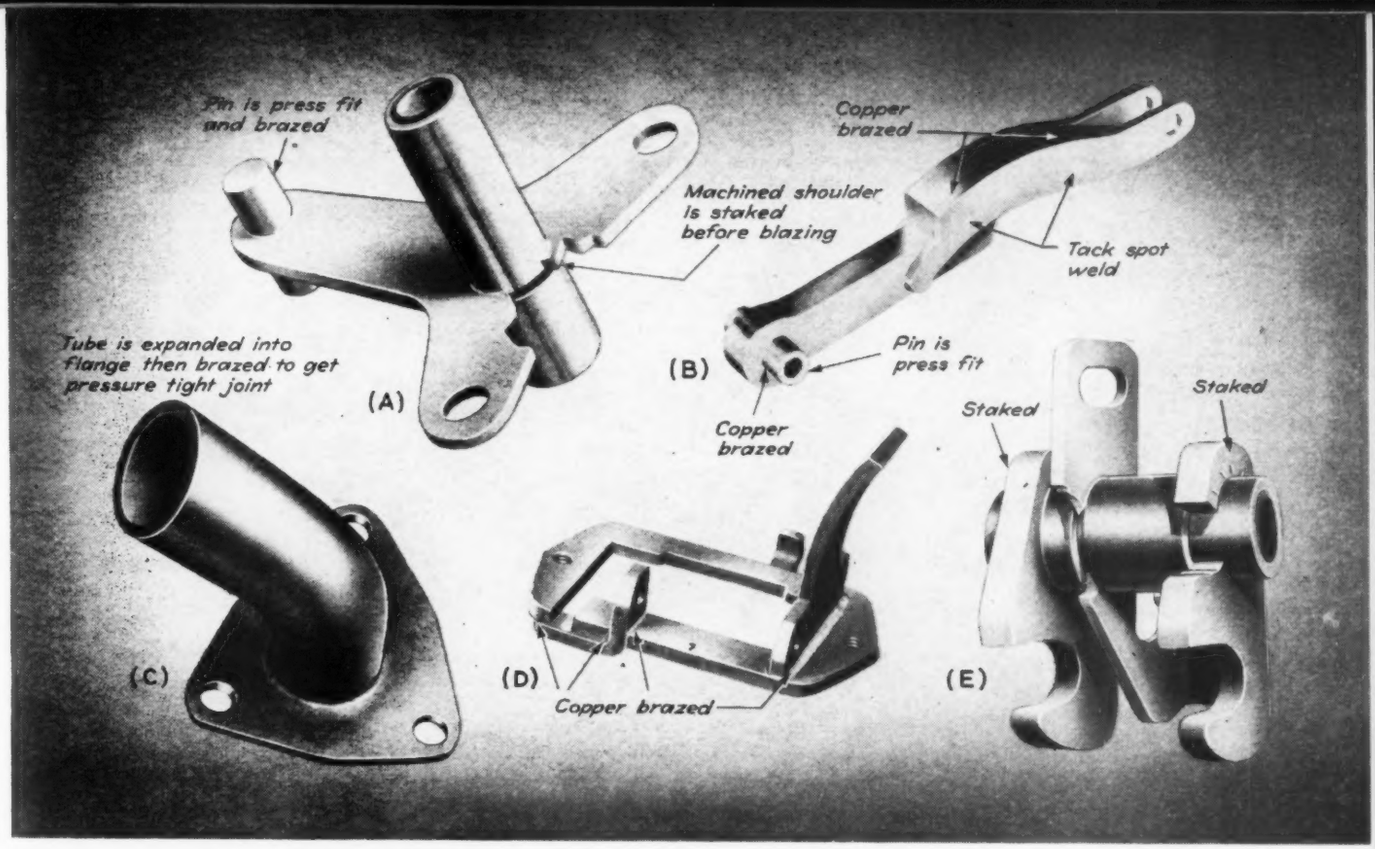


Fig. 1—Copper brazing can be used to join two pieces, or an assembly of a dozen pieces and more, into a composite part by a single brazing operation in a furnace. Relatively simple parts are illustrated by (A) to (E), inclusive.

Copper Furnace Brazed Parts And Principles of Design

KENNETH SCHMIDT
Production Design Engineer, North American Aviation, Inc.

Positive information for guidance in the design of parts to be fabricated by copper brazing, including details for placing the copper and the degree of tightness of joint for best results. The importance of venting and types of vents are discussed. Comparative data on costs are also included.

COPPER BRAZING is a simple, rapid and economical means of permanently uniting metal parts in the fabrication of light and cheap assemblies. Sheet metal stampings can be joined with excellent results, often replacing heavy forgings, castings and parts machined from bar stock, designed for other methods of joining. Although examples could be cited in which dozens of stampings are brazed to form a single composite structure, the Crosley automobile engine for example, the discussion that follows treats the design of simple parts. The same design principles apply whether the part is simple or complex, despite experience that complex parts sometimes require research development and trials in production before a wholly satisfactory design is achieved.

Parts to be copper furnace brazed are thoroughly cleaned and then assembled at room temperature. Some form of temporary holding device should be used to prevent relative movement during the brazing operation. The brazing material, usually in the form of wire, though sheet, powder and other forms are available, is placed in or near the joints to be brazed. The parts are then heated within the furnace maintained at a temperature of 2,050 deg. F. and containing a reducing atmosphere. This atmosphere prevents the metals from oxidizing and frees them from oxides, if present, thereby preparing the surfaces to be wetted by the molten copper. When the brazing material melts, it creeps on the surfaces and is drawn into the joints by capillary attraction, forming an alloy with the body material.

After the parts have been brought up to temperature and held for the proper length of time, they are moved to an adjoining cooling chamber, also containing a controlled atmosphere. The bonds develop great strength upon solidifying. The part is removed from the cooling chamber when it has cooled to a temperature below which contact with the outside air will not

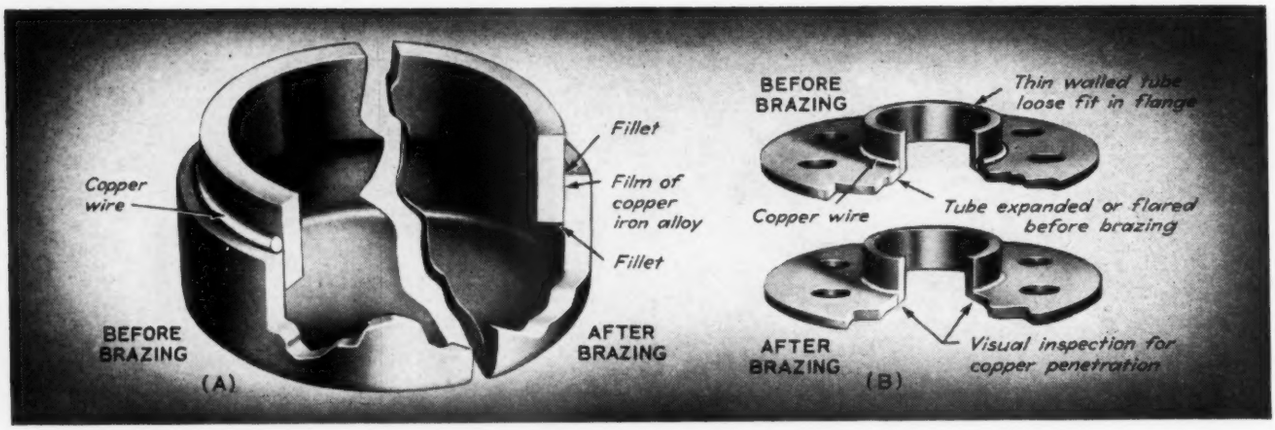


Fig. 2—Typical parts showing how joints should be prepared for brazing and how they should appear afterward.

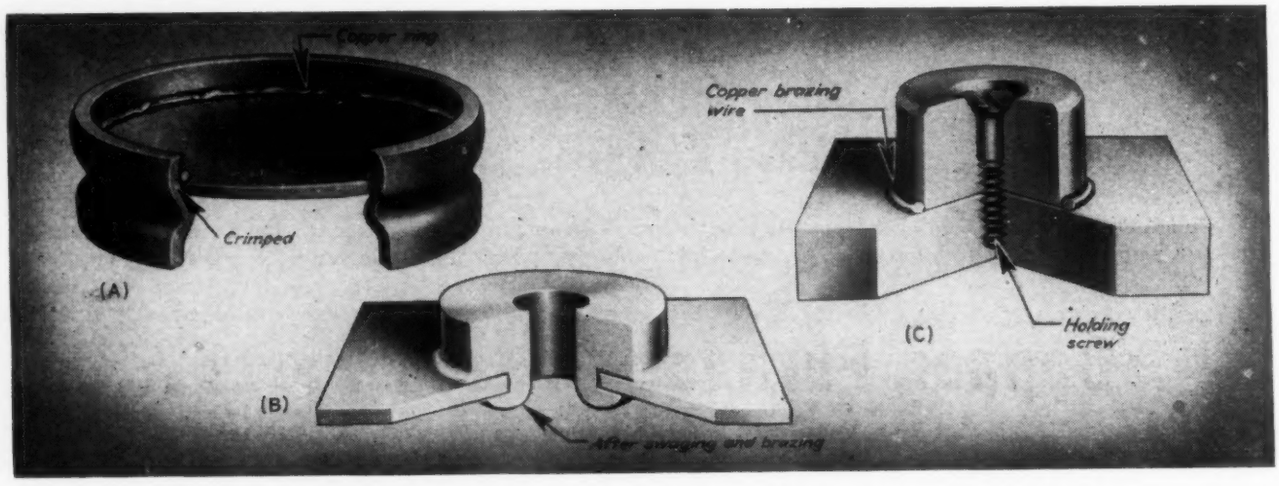


Fig. 3—Some methods of holding the pieces together after assembly and during brazing.

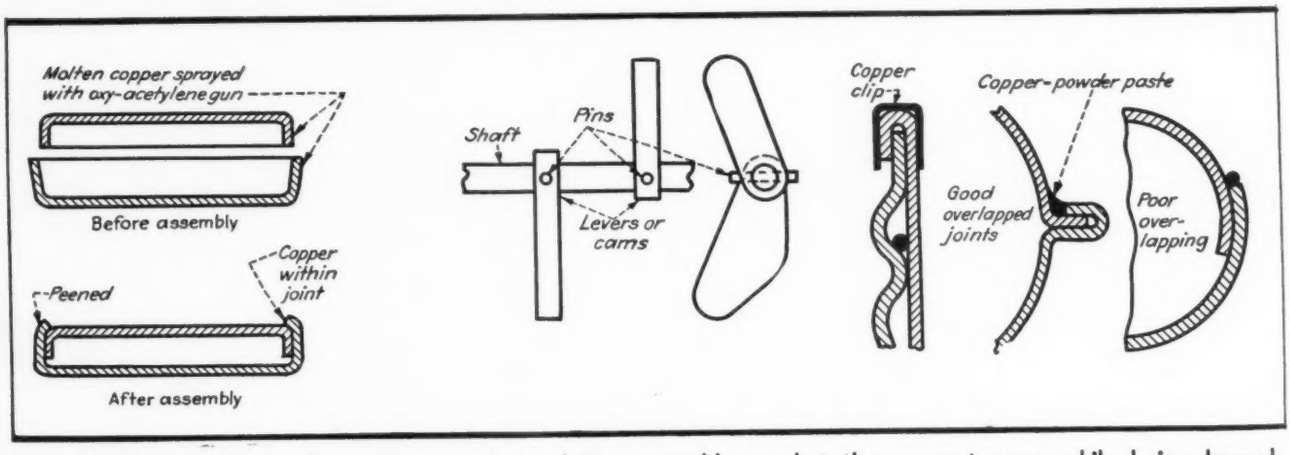


Fig. 4—Other methods of securing the pieces during assembly so that they cannot move while being brazed.

cause discoloration. Thus the part is delivered from the furnace with strong, tight joints and with clean, bright surfaces.

Design Fundamentals

Parts that fit uniformly and snugly over the areas to be brazed are essential to good joints. Interference, snug and press fits are terms of the designer frequently used to classify the type of fit required. Brazing materials, unlike weld metal, cannot be used satisfactorily to fill V-shaped openings between pieces and to join parts having line contact. A maximum clearance of 0.003 in. between mating surfaces is recommended for highest strength joints. Clearances greater than 0.003 in., and sometimes as high as 0.010 in., are used where strength is not the prime factor, but the alloying effect between the copper and the steel is greatly reduced. Sometimes the copper will tend to flow through the joint without brazing. A general rule to follow for interference fits is 0.001 in. per inch of diameter, with 0.003 in. as the maximum. Under extreme press fits the parts may be stretched beyond the elastic limit, thereby causing weak and inferior joints.

The telescopic or tube type of joint is

one of the most common and offers the greatest saving when many large areas are joined in one operation. Fig. 2 shows details of joints of this type. Staking parts together to prevent relative movement during brazing is often used, particularly where clearance fits are allowed or where component parts with wide dimensional tolerances are assembled, without selection, to yield fits varying from tight to loose. Other methods used as temporary holding devices are tack-spot welding, tack welding, crimping, swaging, screw-anchored, Fig. 3, peening, pinning, overlapping and interlocking, Fig. 4. Jigs are not practical to hold the work while in the furnace. The number required for continuous operation of the furnace plus the assortment for each type of joint would be enormous. Another disadvantage of jigs is that they do not withstand the intense heat of the brazing furnace and, if used, would require longer heating periods since they would absorb a certain quantity of heat.

Placement of the brazing material before brazing is an important factor in producing good, strong joints. A good percentage of the poorly brazed parts can be attributed to improper placement of copper. Fig. 5 illustrates a good rule to follow in placing the copper; namely, place the brazing material in such a way that penetration of the joint by the copper can be judged by visual inspection after brazing. Fig. 6 shows other ways of placing copper before brazing. Embedding the copper in special grooves machined at predetermined locations has the disadvantages of added cost for the extra machining and the possibility of improper inspection and incompletely brazed joints. In special designs where the length of the joint is too great for capillary attraction to be effective, placing copper in machined grooves is the only alternative, but some form of destructive tests must be made in order to satisfy structural requirements.

A simple and effective way of brazing parts is to lay one on top of the other with the brazing material either between them or wrapped around one of them near the joint. Fig. 5 shows this type of joint. Here gravity causes the flow of copper rather than capillary attraction.

Parts that would be completely sealed by brazing should be vented. If such parts are not vented, there is danger of the trapped air expanding when heated and blowing the pieces apart. Other possibilities are: Escaping air may displace the copper or retard its flow; air pockets or voids may form, causing incomplete brazing; and the surfaces to be brazed may not be contacted by the reducing atmosphere, consequently optimum brazing conditions would not be attained. Fig. 7 shows typical ways of venting parts.

Materials

The steels most widely used for joining by copper brazing are the carbon and alloy steels. For special installations stabilized stainless steel can be copper furnace brazed, but a flux must be used. This requires extra handling and increases the cost of the brazing operation. When combinations of the carbon and alloy steels are brazed together, heat-treatment requirements of only one type can be assured, although they might fall within the ranges of both steels. The parts to be brazed must be clean and free of greases, oils, rust or other scales in order to be wetted by the molten copper. Copper flows best on clean, matte surfaces. Regular machine finishes having surface roughness values of 100 to 250 rms. are

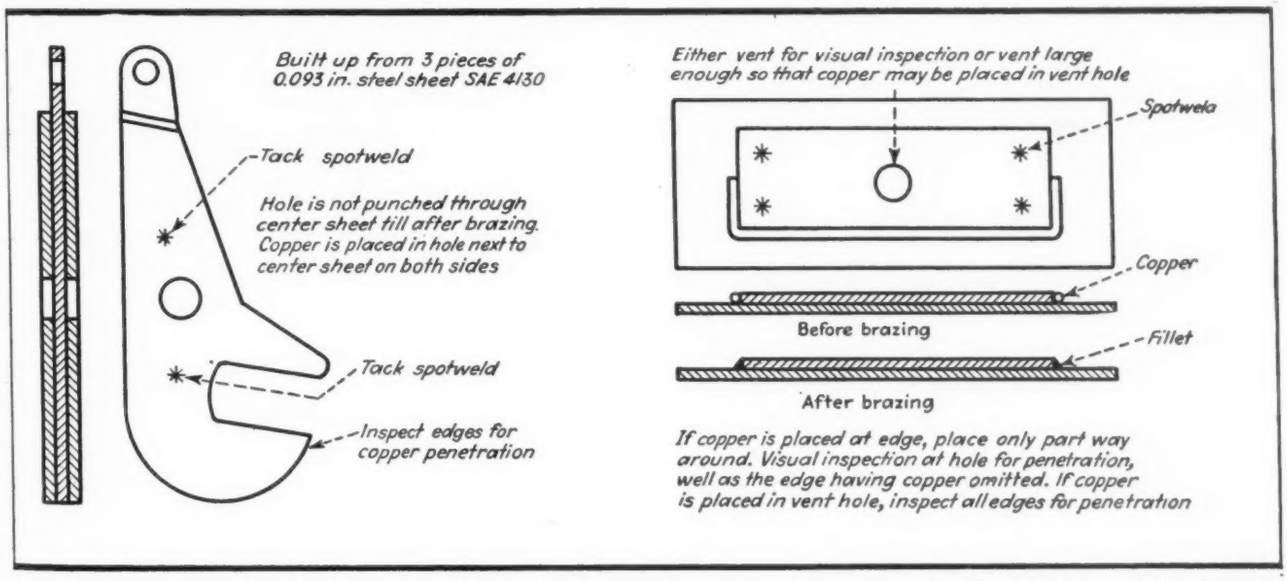

Built up from 3 pieces of 0.093 in. steel sheet SAE 4130
Tack spotweld
Hole is not punched through center sheet till after brazing. Copper is placed in hole next to center sheet on both sides
Tack spotweld
Inspect edges for copper penetration
Either vent for visual inspection or vent large enough so that copper may be placed in vent hole
Spotweld
Copper
Before brazing
Fillet
After brazing
If copper is placed at edge, place only part way around. Visual inspection at hole for penetration, well as the edge having copper omitted. If copper is placed in vent hole, inspect all edges for penetration

Fig. 5—Diagrams with instructions on placing the copper in such a manner that the thoroughness of the brazing operation can be determined by visual inspection. Properly-placed copper is essential to well-brazed joints.

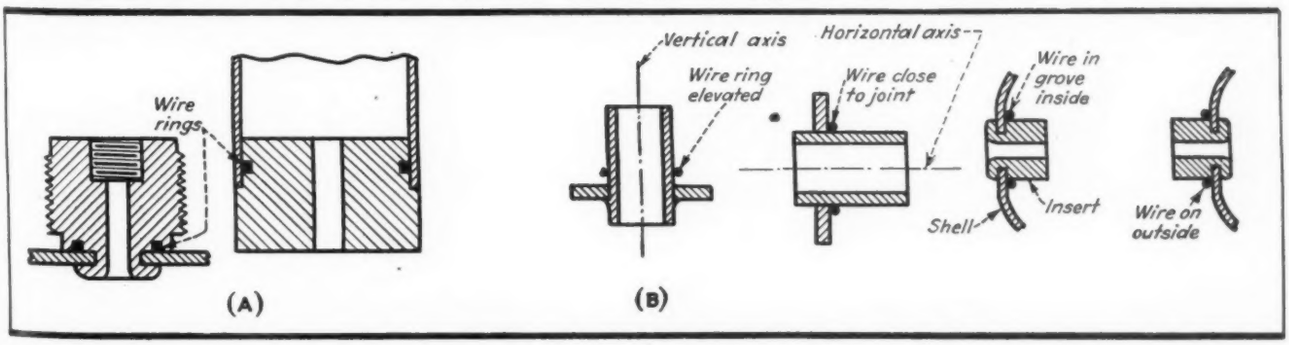


Fig. 6—(A) Sometimes the copper brazing wire is embedded in grooves. (B) More ways of placing the copper.

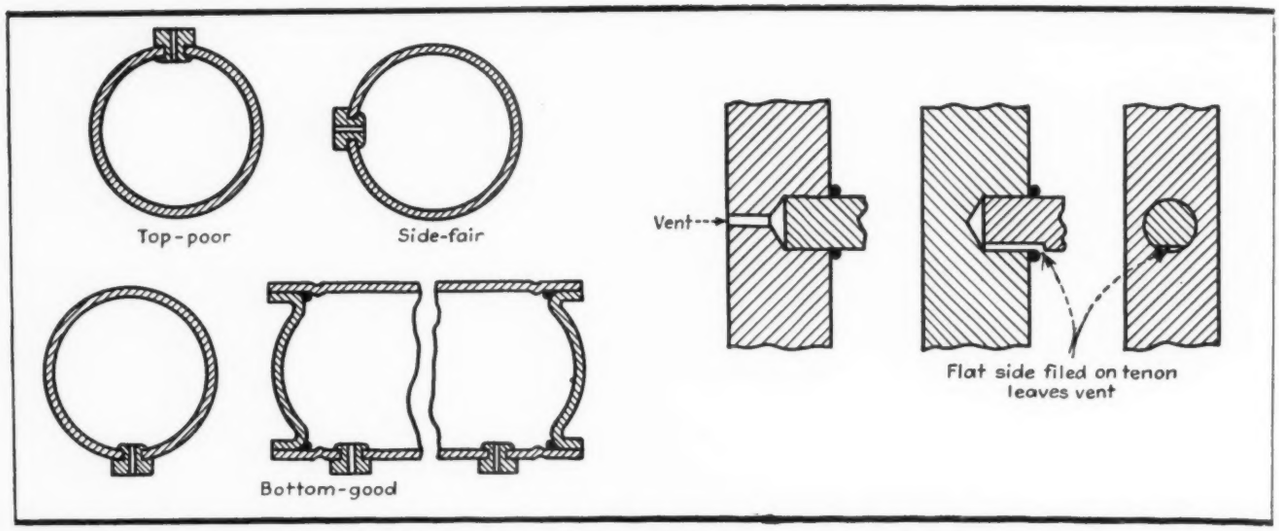


Fig. 7—Vents in the bottom of inclosed spaces allow the lighter furnace gases better to purge the spaces of air and heavy vapors. Entrapped air when heated may blow the pieces apart or retard flow of copper during brazing.

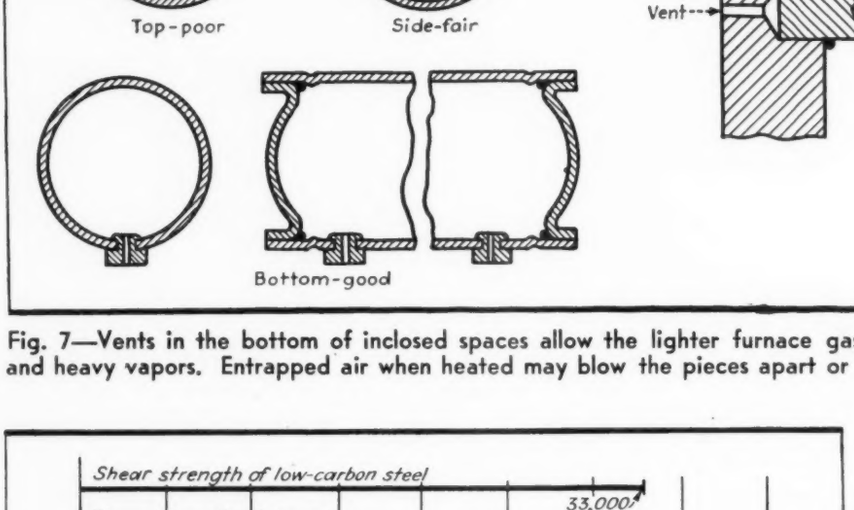


Fig. 8—Results of tests showing relative strength of copper-brazed joints with several classes of fit, compared with the shear strengths of annealed low-carbon steel and of copper.

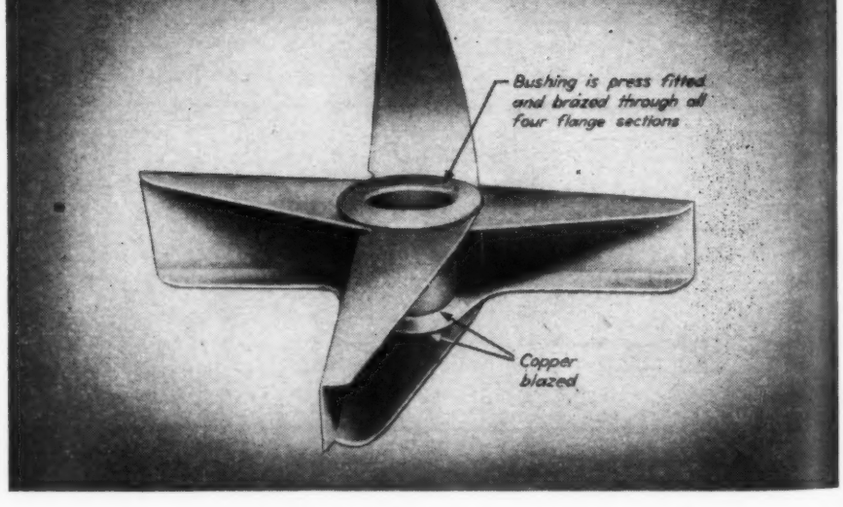
Fig. 9—A tail wheel trunion fitting that cost $2.61 as a forging and $1.87 as a copper brazed assembly.

satisfactory. Highly polished surfaces or ground finishes having values of 40 rms. or better, hinder the wetting action of the copper and sometimes allows the copper to run off the surfaces without any wetting action at all. Shot-blasted and vapor-blasted surfaces are more satisfactory than sand-blasted surfaces because the sand particles remain embedded in the steel surface and repel the copper, reducing wetting action.

Copper brazed joints are designed for sheer loading only, using an allowable shear strength of 10,000 lb. per sq. in. as given in ANC-5, "Strength of Aircraft Elements." The only military specification that covers copper furnace brazing for aircraft parts is AAF Specification 20018. Although the shear strength of copper is approximately 23,000 lb. per sq. in., the alloy formed between the copper and steel increases the joint strength considerably. Ultimate shear strengths in the neighborhood of 40,000 lb. per sq. in. are consistently obtained in production. Figs. 8 and 10 show comparative test results of copper brazed joints for various diametrical fits and different materials.

Treatment After Brazing

Parts emerge from the cooling chamber with clean, bright surfaces and in the annealed or slightly normalized condition.



The temperature in the furnace removes any work-hardening gained during fabricating processes. If heat-treatable steels do not require any subsequent heat-treatment or casehardening after brazing, they are usually normalized to relieve strains and improve their grain structure.

Heat-treating and casehardening are done after brazing. Steels can be heat-treated to specific requirements, corresponding to the type of steel used, as shown in Fig. 10. Excess copper must be removed from the surfaces before hardening. Pickling, sandblasting, deplating and stripping processes can be used for removing excess copper.

Copper brazed assemblies can be successfully cadmium or chrome plated, but when hard chrome plating is required, the surfaces must be stripped of copper before plating.

In-line drilling, milling, turning, grinding and other finish machining operations are done after brazing or heat-treatment in order to meet dimensional tolerances. Mismatch along edges of laminated parts can be removed, holes can be aligned, cylindrical surfaces can be ground to the desired degree of smoothness and any distortions developed from heat-treating can be removed by these finish machining operations. Surface finishes better than 100 rms. cannot be guaranteed if parts are heat-treated after brazing in furnaces not containing controlled atmospheres.

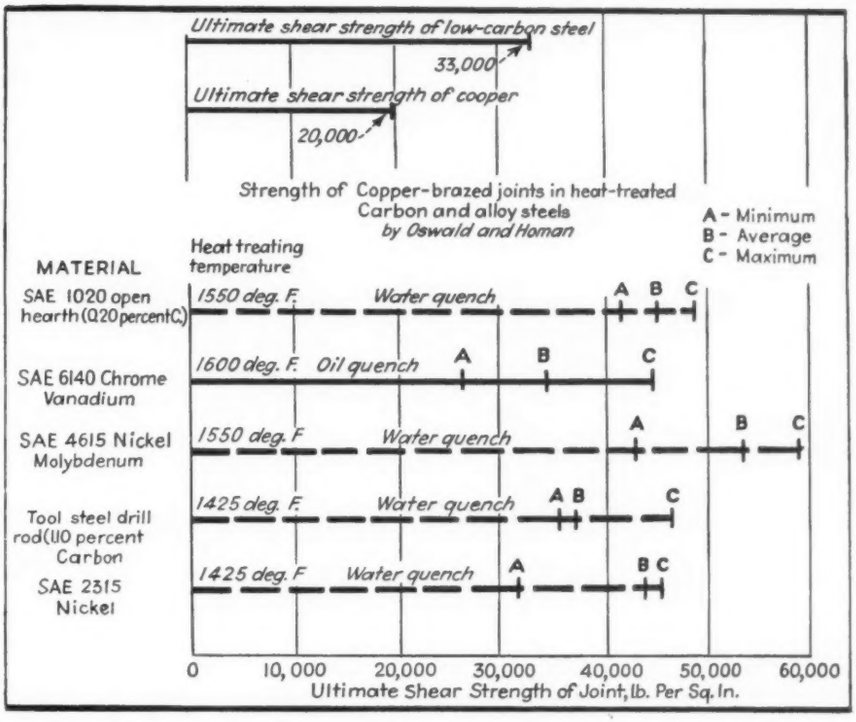


Fig. 10—Carbon and alloy steels, copper brazed and heat-treated, show good strength in brazed joints, which accounts for their general use.

Economy of Manufacture

On the other hand, the designer should be cautioned about adding unnecessary machining. There is little to gain from joining by copper brazing unless parts are produced with savings in time, effort or cost. Since these savings are effected largely through the stamping, punching, drawing and pressing of parts to the desired shapes with a corresponding reduction in material saving and the amount of machining, the designer can well afford to study the need for every machining operation.

A few examples describe better than words how machining can be obviated or reduced to obtain saving in cost. Fig. 11 shows a part that was made not only at less cost but also with a saving in weight, both items being important in the design of aircraft parts.

The foregoing comparisons pertain to volume production, but copper brazing can be economical even in building experimental models and parts in small quantities. A part can be compounded from machined pieces as well as from stampings. The machined pieces can be simple and they need not be identical with stampings that would serve the same purpose.

The cost of furnace brazing depends upon the assembly time of the component parts, the size and weight of the assembly, the quantity of filler material required and the time in the furnace. Large parts are more expensive in proportion to small parts because they not only take more space in the furnace but also require longer heating periods to raise them to the brazing temperature. For economy in the furnace, small parts having many joints are preferable to large parts with few joints.

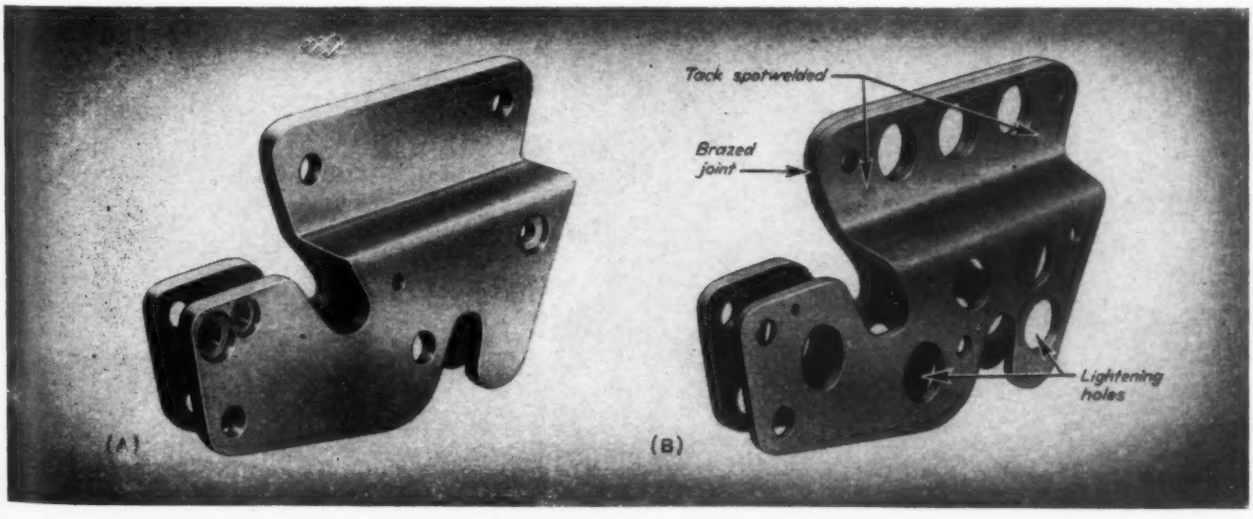


Fig. 11—(A) Part machined from bar stock cost $2.25; (B) Steel sheet pieces blanked and brazed together for $1.25.

TYPICAL METHODS OF SEALING

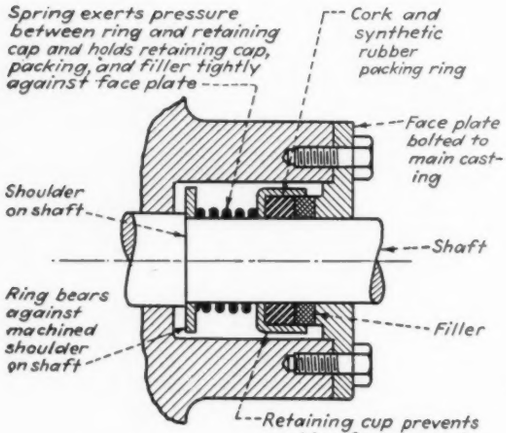

Spring exerts pressure between ring and retaining cap and holds retaining cap, packing, and filler tightly against face plate
Cork and synthetic rubber packing ring
Face plate bolted to main casting
Shoulder on shaft
Shaft
Ring bears against machined shoulder on shaft
Filler
Retaining cup prevents packing from expanding

FIG. 1

Armstrong Cork Co.

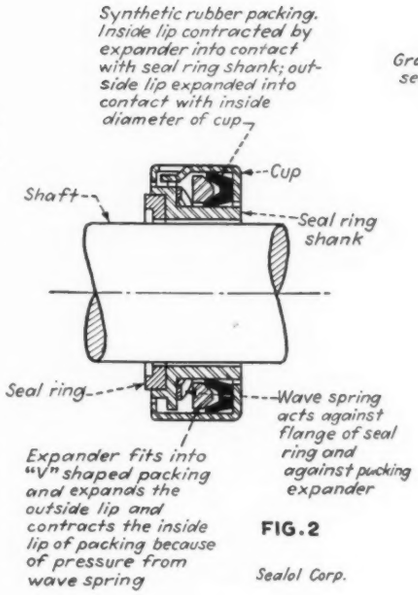

Synthetic rubber packing. Inside lip contracted by expander into contact with seal ring shank; outside lip expanded into contact with inside diameter of cup
Cup
Shaft
Seal ring shank
Seal ring
Wave spring acts against flange of seal ring and against packing expander
Expander fits into "V" shaped packing and expands the outside lip and contracts the inside lip of packing because of pressure from wave spring

FIG. 2

Sealol Corp.

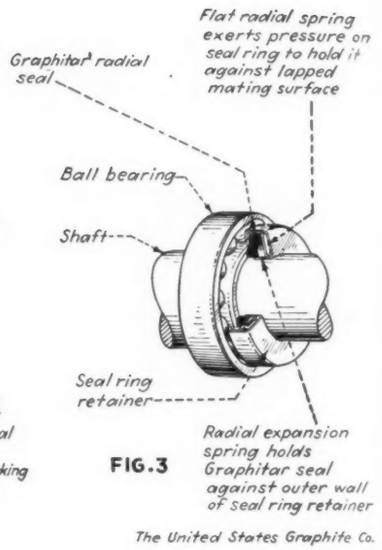

Flat radial spring exerts pressure on seal ring to hold it against lapped mating surface
Graphitar radial seal
Ball bearing
Shaft
Seal ring retainer
Radial expansion spring holds Graphitar seal against outer wall of seal ring retainer

FIG. 3

The United States Graphite Co.

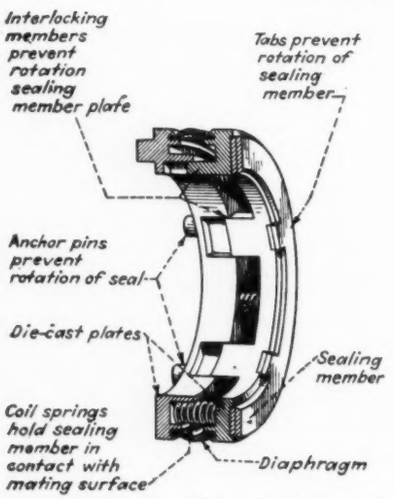

Interlocking members prevent rotation sealing member plate
Tabs prevent rotation of sealing member
Anchor pins prevent rotation of seal
Die-cast plates
Sealing member
Coil springs hold sealing member in contact with mating surface
Diaphragm

FIG. 4

Chicago Rawhide Mfg. Co.

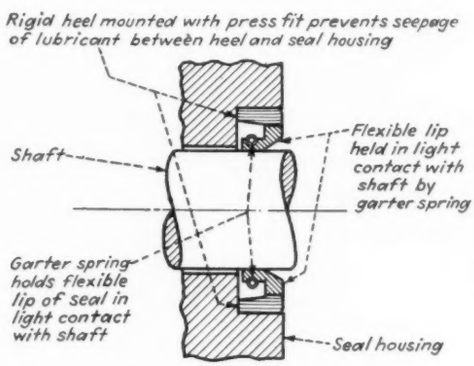

Rigid heel mounted with press fit prevents seepage of lubricant between heel and seal housing
Shaft
Flexible lip held in light contact with shaft by garter spring
Garter spring holds flexible lip of seal in light contact with shaft
Seal housing

FIG. 5

Johns-Manville, Inc.

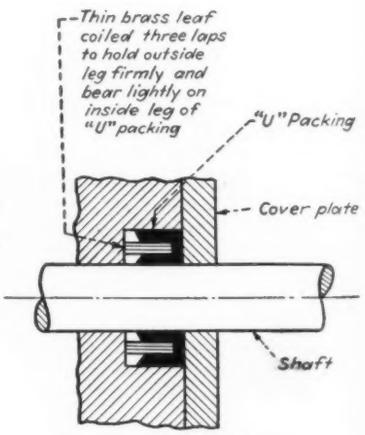

Thin brass leaf coiled three laps to hold outside leg firmly and bear lightly on inside leg of "U" packing
"U" Packing
Cover plate
Shaft

FIG. 6

Chicago Belting Co.

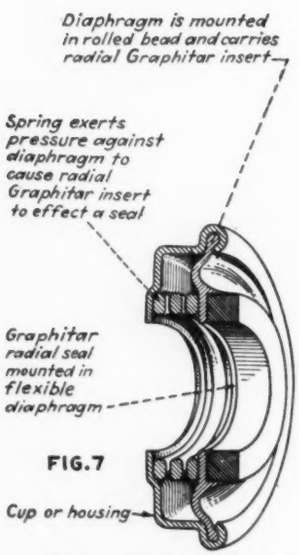

Diaphragm is mounted in rolled bead and carries radial Graphitar insert
Spring exerts pressure against diaphragm to cause radial Graphitar insert to effect a seal
Graphitar radial seal mounted in flexible diaphragm
Cup or housing

FIG. 7

The United States Graphite Co.

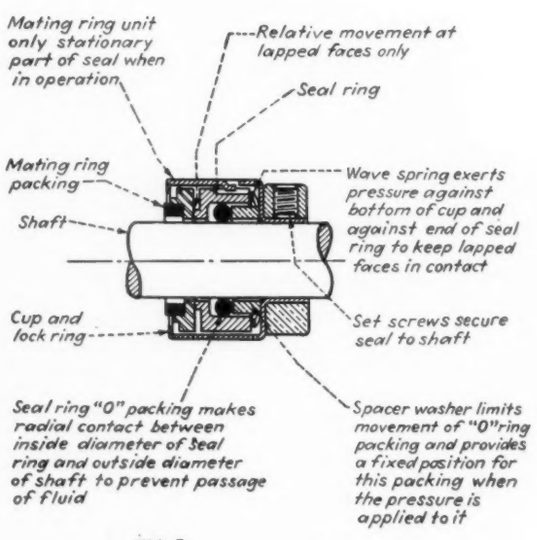

Mating ring unit only stationary part of seal when in operation
Relative movement at lapped faces only
Seal ring
Mating ring packing
Wave spring exerts pressure against bottom of cup and against end of seal ring to keep lapped faces in contact
Shaft
Cup and lock ring
Set screws secure seal to shaft
Seal ring "O" packing makes radial contact between inside diameter of seal ring and outside diameter of shaft to prevent passage of fluid
Spacer washer limits movement of "O" ring packing and provides a fixed position for this packing when the pressure is applied to it

FIG. 8

Sealol Corp.

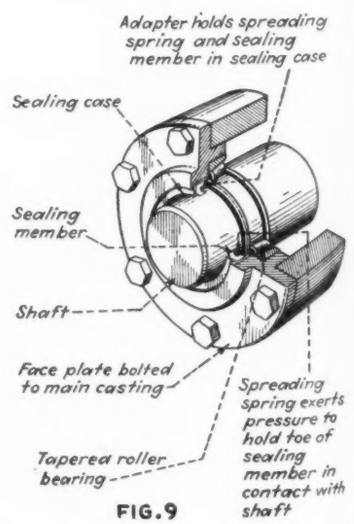

Adapter holds spreading spring and sealing member in sealing case
Sealing case
Sealing member
Shaft
Face plate bolted to main casting
Tapered roller bearing
Spreading spring exerts pressure to hold toe of sealing member in contact with shaft

FIG. 9

ROTATING SHAFTS

Shaft seals, used to prevent seepage of lubricating oils or other liquids, range from simple flange packings to elaborate bellows and lapped-surface devices. Some of these, utilizing wave, coil, and garter springs, diaphragms, and locking rings, are described and illustrated.

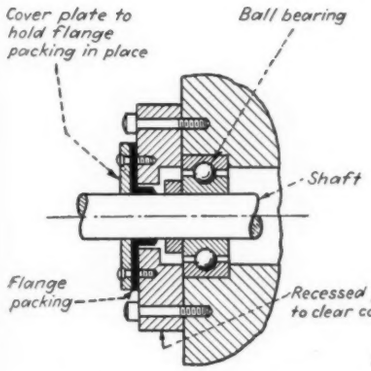


FIG. 10 Chicago Belting Co.

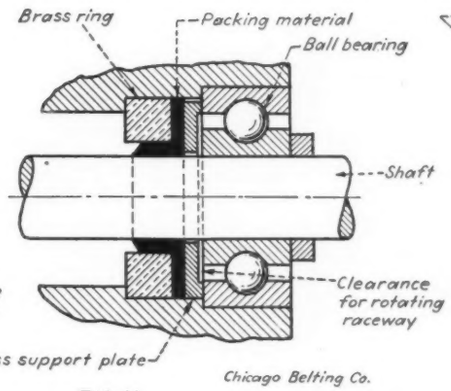


FIG. 11 Chicago Belting Co.

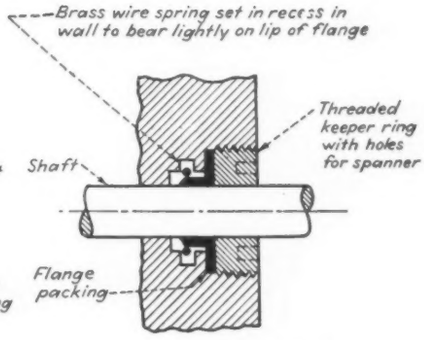

Brass wire spring set in recess in wall to bear lightly on lip of flange
Threaded keeper ring with holes for spanner
Shaft
Flange packing

FIG. 12 Chicago Belting Co.

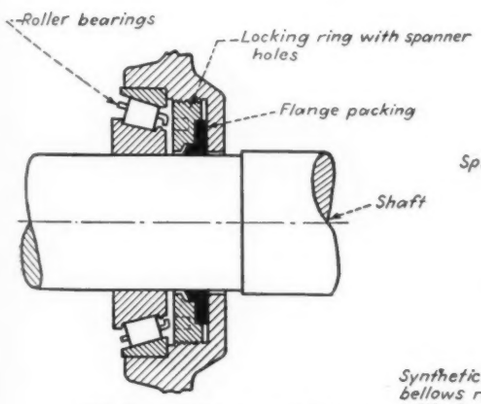


FIG. 13 Chicago Belting Co.

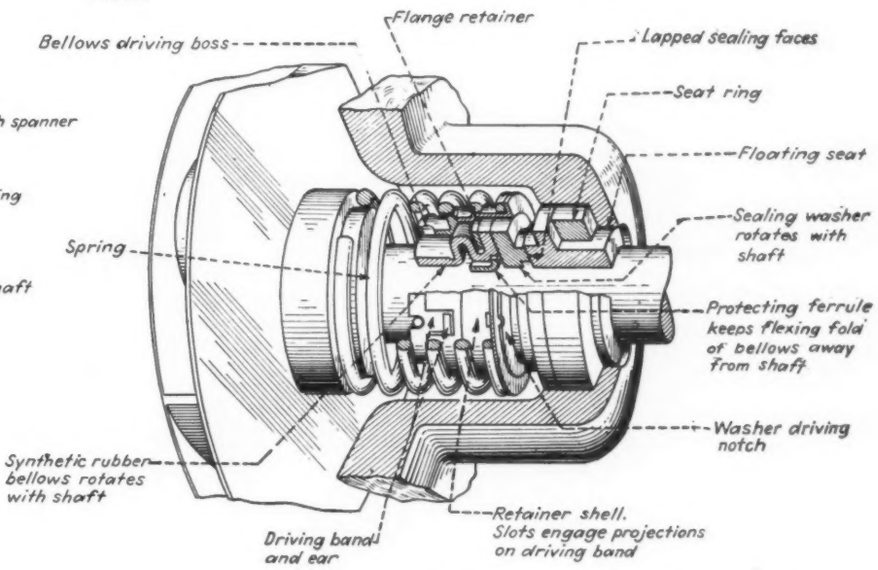

Flange retainer
Bellows driving boss
Lapped sealing faces
Seat ring
Floating seat
Sealing washer rotates with shaft
Spring
Protecting ferrule keeps flexing fold of bellows away from shaft
Washer driving notch
Synthetic rubber bellows rotates with shaft
Retainer shell. Slots engage projections on driving band
Driving band and ear

FIG. 14 Crane Packing Co.

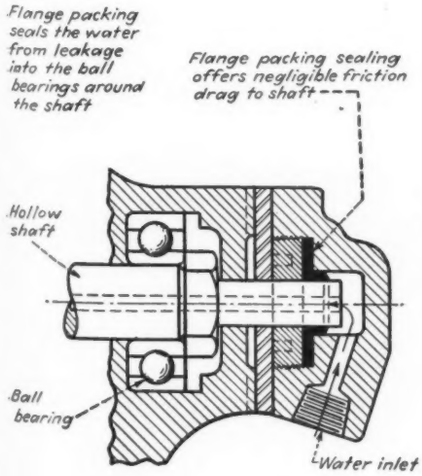
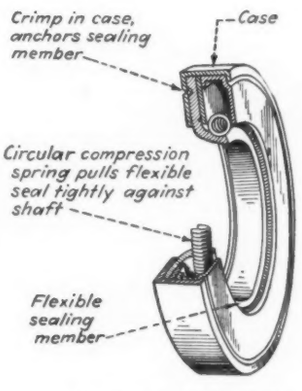

Flange packing seals the water from leakage into the ball bearings around the shaft
Flange packing sealing offers negligible friction drag to shaft
Hollow shaft
Ball bearing
Water inlet

FIG. 15 Chicago Belting Co.


Crimp in case, anchors sealing member
Case
Circular compression spring pulls flexible seal tightly against shaft
Flexible sealing member

National Motor Bearing Co. Inc.

FIG. 16

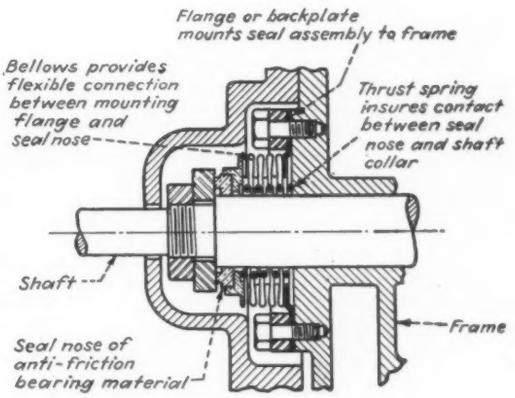

Flange or backplate mounts seal assembly to frame
Bellows provides flexible connection between mounting flange and seal nose
Thrust spring insures contact between seal nose and shaft collar
Shaft
Frame
Seal nose of anti-friction bearing material

FIG. 17 The Fulton Sylphon Co.

Engineering Management Fundamentals Adapted to the Small Company

JAMES E. THOMPSON
Chief Production Engineer, Parkhurst & Company

Organization for the efficient scheduling and controlling of the flow of work in engineering departments. Six essential principles of engineering management and their practical application are discussed. The advantages derived from maintaining complete personnel experience records are outlined.

EFFICIENT engineering management depends upon the application of six fundamentals: (1) A clearly defined objective; (2) a detailed plan for reaching the objective; (3) a simple, practical organization; (4) competent personnel; (5) control of the work flow; and (6) adequate follow-up. When these fundamentals are correctly understood and applied, each person having managerial or supervisory duties will have control over his objective, plan, organization, men, methods and schedules.

The most important fundamental is maintaining a clear-cut understanding of the job to be done—both today, and as far into the future as practicable. All administrative and supervisory personnel must constantly have a complete understanding of the over-all, long-range engineering program. Each must understand the exact nature of his responsibility for carrying out the program. The degree of an individual's knowledge of details should increase, and the scope of his responsibility decrease in proportion to the nearness of the working level.

It is always desirable that everyone in the engineering department know the general aspects of the engineering program. Departmental morale and efficiency suffer severely whenever personnel are excluded from knowledge of the general nature and purpose of the work in process. Only rarely do security considerations make it necessary to restrict such information.

Too much effort in planning the course to be followed in reaching the objective of an engineering program is hardly possible. This planning must be continuous, for an engineering program does not exist simply for consummation of one project. It is constantly in the process of revision, expansion and contraction as projects are completed and new work planned.

Once a plan is prepared its successful fulfillment becomes largely a matter of delegating each piece or parcel thereof to competent personnel, followed by proper coordination and control to insure successful achievement within alloted times. Establishment of a detail plan enables accurate cost estimating and successful cost control. The detail plan of each project should be prepared in a form that permits issuing copies to each person concerned with its accomplishment. This information should be revised and reissued whenever a significant change takes place.

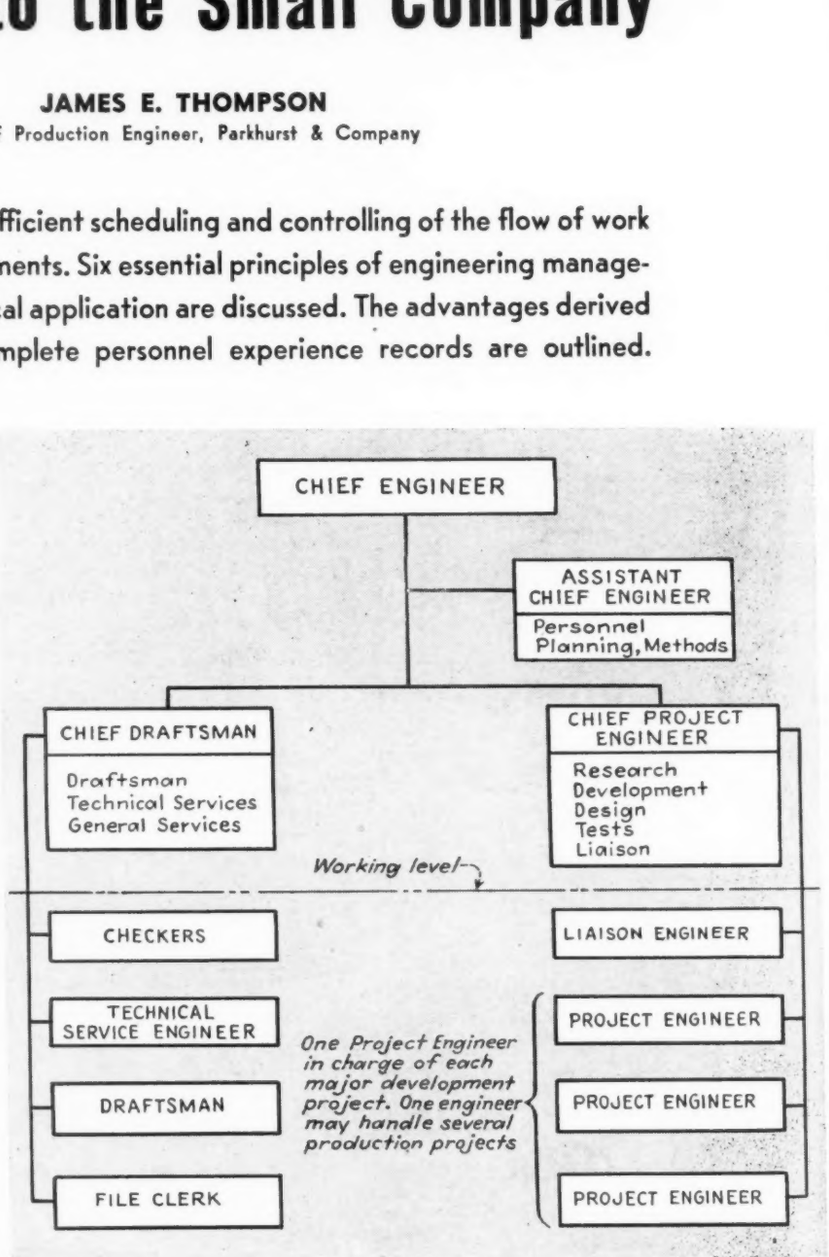


Fig. 1—Functional organization chart for a small engineering department. Chart is readily adaptable to consolidation to suit the needs of small departments or expansion to fill the requirements of departments of moderate size.

Use of a design information report for each project insures that all personnel will work to identical basic design data; and precludes the possibility of time being wasted through misunderstanding of the course to be followed during development of the design. It will further insure that only those designs and purchased parts approved by customer and engineering man-

agement will be incorporated into the project.

Methods of collecting and allocating costs are a basic part of the plan for managing an engineering department, since intelligent planning is impossible without accurate knowledge of costs. Cost control should go beyond the strictly timekeeping and accounting phases of collecting and allocating costs. It should also encompass establishment of departmental budgets and cost standards.

The first step is an over-all budget for the engineering department, followed by individual budgets for each operating section. These budgets must have flexibility for possible revision to accommodate changes in engineering plan. Cost standards should be established for all repetitive work, such as drawing release, blueprinting and parts listing. In these operations the time expended is directly proportional to either the quantity or area of drawings processed. Budgets can be established and revised on the basis of estimated drawing quantities. Actual efficiency can be measured as the percentage realization of the standards, and can be increased by reducing all unnecessary operations.

Budgets for design and drafting work are difficult to reduce to standards. While drawing quantity and area are still measures of the work done, the effort required will be proportional to the complexity of the design. Only by comparison with the cost of similar engineering work, performed under similar conditions, is it possible to arrive at reasonably accurate design and drafting estimates. All estimates and budgets presented to top management should be realistic, containing suitable allowances for all reasonable contingencies. Nothing discredits the engineering department more rapidly than repeated failure either to meet completion dates or to maintain budgets.

Once budgets are established it is necessary that each section of engineering operate within its current allowance, and be guided by periodic cost reports. A time-change system should be established to provide daily cost accumulation by charging each engineering hour against specific, designated account numbers.

Organization

Departmental organization can take the form indicated by the chart, Fig. 1. In small departments the organization will be abridged but identical in principle. There will be no full-time administrative or service functions. Instead, each draftsman and engineer will be assigned a portion of the service duties found in a large department. These can be discharged without seriously hampering the employee's main assignment of drafting or design, as the non-drafting demand upon his time will be relatively small. For instance, one draftsman can be assigned part-time duties as file clerk and blueprint operator. Another can devote some time to liaison between engineering and the factory departments. Still another may spend time preparing handbooks, specifications, service bulletins, and the like. The net result will be the assignment of all important technical and general service functions as part-time duties of specified personnel and without hampering their prime purposes of design and drafting. Each will report directly to the chief engineer, and in a sense will serve as both supervisor and worker.

It may be desirable also to establish the position of assistant chief engineer, particularly for a department of intermediate size (between twenty and fifty employees), for the purpose of handling personnel, planning and methods, as well as providing for a department head during absence of the chief engineer. If this position is not considered desirable, then personnel, planning and methods may be handled by the chief draftsman.

Personnel Control

Every engineering department, irrespective of size, requires personnel control. This requirement becomes more noticeable as the department increases in size, until a point is reached (approximately 300 employees) where personnel matters require the full-time attention of an engineering personnel manager.

Certain personnel control procedure is necessary in every department, and involves simple, methods of handling personnel procurement and termination, maintaining assignment and history records, wage and salary reviews, and overtime authorization. In a small organization the recording and preliminary screening in personnel procurement can be handled by the company personnel department.

It is always desirable to maintain complete employee history records, which show education, experience and other particulars regarding each engineering employee. When this information is readily available to the chief engineer, considerable time can be saved when planning new job assignments, promotions and wage adjustments.

A definite policy of wage and salary reviews should be established, as a practice of hit-or-miss increases cannot be tolerated. At specified periods all employees should be reviewed on the basis of job performance to ascertain if an increase in wage or salary is merited. All job performance reviews should be made by the employee's supervisor, subject to the approval of the chief engineer.

Overtime is an evil that should be avoided wherever possible, and under no circumstances should it become customary. When overtime work exceeds a few consecutive days, the continued mental fatigue causes employees to slow down to a point where the overtime does not increase output. Quality of work rapidly deteriorates when employees are subjected to a continued overtime schedule.

There are occasions where overtime is necessary, but all such work should receive the chief engineer's approval, which should be given only where there is no other possible manner of getting the required work done. A definite procedure for requesting and authorizing overtime is necessary when the department is large enough to require full-time supervisors.

Another problem in every department is control of employees who must visit factory departments in connection with engineering work. The prerogative of freely visiting other departments cannot be extended to all engineering employees, for there are always a few who will take advantage of the privilege. In a large department it is feasible to inaugurate a

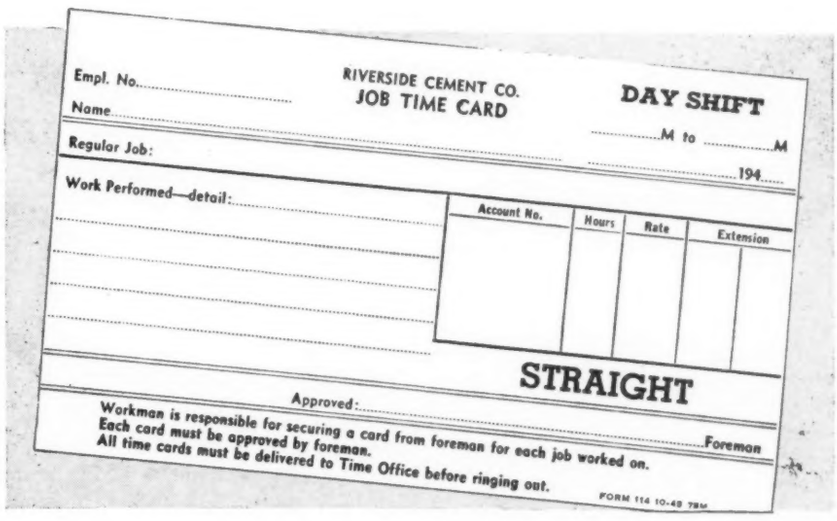


Fig. 2—A time-charge card such as this can be used in making a daily report of the time of an employee to be charged against each specific job.

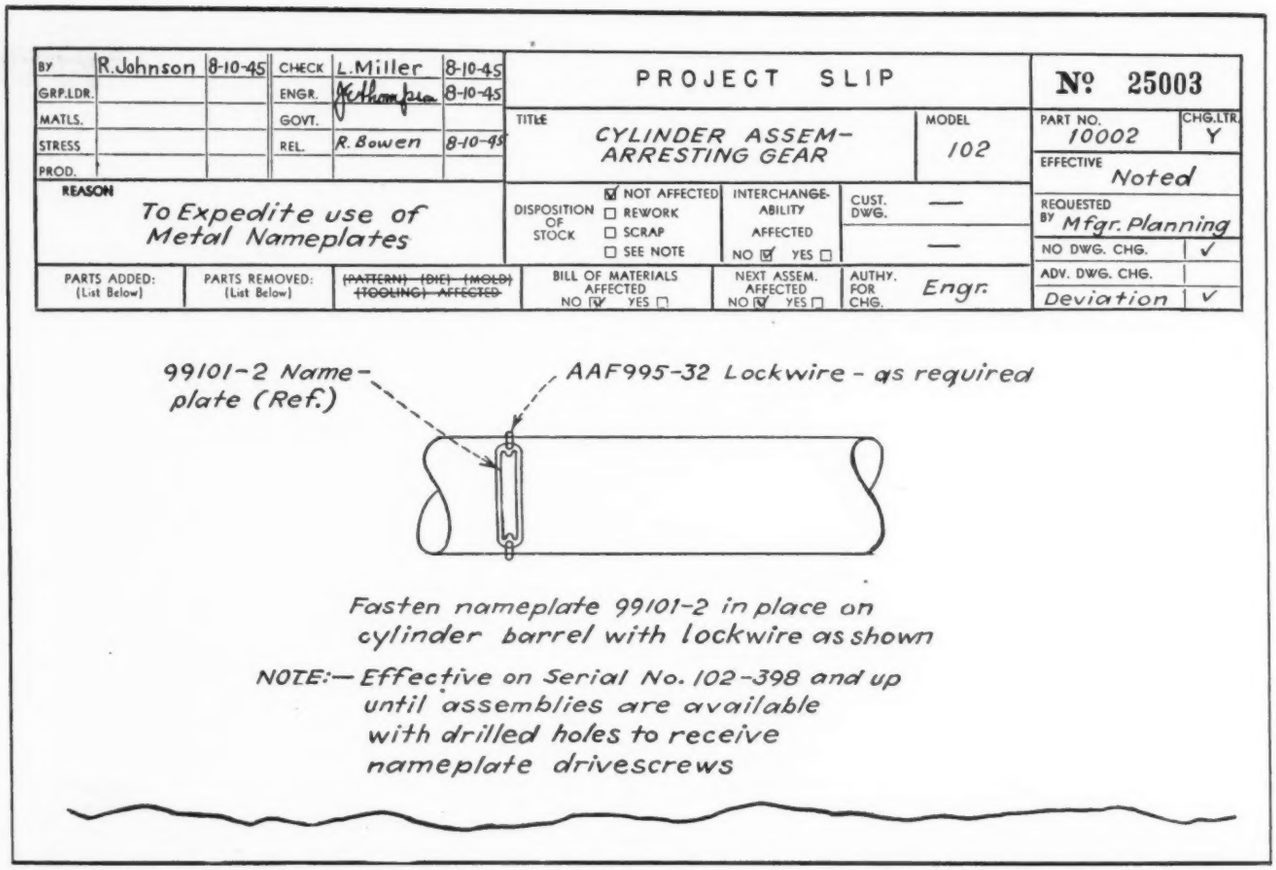


Fig. 3—Project slip, on an 8½ x 11 in. reproducible form, when checked in the appropriate boxes and properly titled, is suitable for issuance of all types of advance information by small engineering departments.

shop-pass system. In small departments it usually suffices to require that all employees leaving the engineering department sign a register noting the time of their departure and destination, followed by entering the time of their return. This not only tends to eliminate extended visits to other departments, but also makes it possible to page engineering employees who are visiting other departments.

Controlling the Work Flow

Basic control of the work flowing through the engineering department is a responsibility of the engineering planning activity, who schedule the amount of work in process at any given time. Detail control involves establishing methods to control: (1) Preparation of drawings; (2) drawing numbers; (3) drawing releases; (4) distribution of engineering data; (5) advance information; (6) drawing changes; (7) master parts lists, and (8) liaison with factory. These are the major factors in the physical preparation and release of drawing information, and are the engineering activities most apparent to other departments.

In every engineering department, irrespective of size, there is need for standardizing the preparation of drawings. The larger engineering departments usually issue a drafting room manual or engineering manual to every new employee. This publication defines and illustrates the exact manner of preparing drawings that will be acceptable to the checkers.

Small engineering groups do not justify the expense of preparing an elaborate manual, but the basic requirements for drawing and preparation still need to be clearly defined in instructions to every draftsman. These data may comprise only a few mimeographed sheets, but will be effective if they define the department's standard organization; drawing sizes, scales, titles and notes; methods of specifying materials, dimensioning, and calling-out parts requirements; and procedure for releasing drawings. A good method of presenting them to a small engineering group is through brief printed instructions, supplemented with a few well chosen specimen drawings that illustrates common conditions encountered during preparation of drawings.

To establish and maintain a master numbers book is vitally necessary. Recording the numbers of all drawings in the book should be made the responsibility of some one member of the department. Under no circumstances should the numbers book be available to the draftsmen, with dependence placed upon each employee for making proper entries in the book upon using the next available number. The latter practice is certain to lead to costly errors, arising from failure to record numbers and use of the identical number on more than one drawing. Centralized control of drawing numbers is even more important when the numbering system is "coded," that is, the number indicates the nature of the part shown on the drawing. Unless rigid control is maintained over issuance of numbers in a coded system, confusion is bound to follow.

Drawing Release

The exact nature of all drawings that have been completed, checked, approved and released by the engineering department is essential information. These items must be listed by number, date and revision, in a manner that provides for rapid reference. The drawing release system should provide: (1) Authority for releasing the drawing; (2) design approval of the drawing; (3) record of all change requests and advance engineering information incorporated in each change; (4) record by change letter and date of each release of the drawing; (5) record of the serial number and destination of each

print made of the released drawing; and (6) delivery acknowledgment from each recipient of prints.

A positive control must be established over the release of engineering data to avoid the possibility of other departments working to obsolete drawings or specifications. All drawing and report revisions should be identified by change letters or revision numbers (preferably the former) and a procedure established for automatically supplying a copy of each revision to all holders of previous releases. The importance of maintaining rigid control over the release of engineering data cannot be overemphasized, as manufacturing parts to obsolete drawings can be extremely costly.

All copies of drawings, specifications and reports issued by the engineering department should be provided with individual identity. This can be easily accomplished by assigning to each a serial number providing both numerical identification and record of the revision letter.

A record of the distribution of engineering data must be maintained, showing for each drawing, report and specification the date of the original distribution and the destination of each copy, as well as any subsequent distribution of copies of revised issues. A shipping notice or invoice should accompany each shipment of engineering data to every destination, prepared in duplicate with one copy to be signed by the recipient and returned as acknowledgment of delivery. This practice should be followed both for data delivered to other departments within the company, and to destinations outside of the company.

Advance Information

Issuance of engineering information in advance of actual drawings can lead to endless confusion and misunderstanding unless properly controlled and authorized. When properly handled this practice can effect large savings by expediting construction of experimental articles and facilitating changes during production. Control is readily established by issuing advance information in a standardized manner, subject to definite centralized approval, and with complete records of its distribution.

Small engineering departments have correspondingly little need for advance drawing changes. The majority of drawing changes can be quickly made, followed by rapid distribution of revised prints. Also, need for stop orders will be infrequent. Usually a small department can use a single form for issuance of all types of advance information—similar to the "Project Slip" shown in Fig. 3.

Project Slips are prepared in the same manner as advance drawing changes or engineering orders. The title block provides for indicating the purpose of the project slip (PS). When the "No Dwg Chg" space is checked the PS is likely to be a drawing deviation or special test instructions, and will not be incorporated into a drawing. When the "Adv Dwg Chg" space is checked, the PS is an advance drawing change, and the information will be incorporated during the next drawing revision. The PS form is also used as a stop order, stop order release, and salvage order by entering the appropriate title in the blank space at the lower, right corner of the title-block.

Drawing changes are a normal portion of engineering department operation, and it is necessary to establish orderly methods for requesting, making and recording these revisions. The first requirement is a formal method for other departments to request drawing changes—one that will insure proper action in all cases. The second is a logical, efficient system of preparing and identifying drawing revisions.

A drawing change-request form should be provided—even in the smallest organization. This need not be elaborate, and its basic requirements are: (1) Identification of the pertinent drawing; (2) statements of the change desired and the reason for the request; (3) an identifying serial number for the change request; (4) a copy to be retained by the originator; and (5) a copy to be returned to the originator with notation of engineering action.

Use of change request serial numbers is important. Consistent, rapid action on requested changes is dependent upon providing individual identity for each request. It is then a simple matter to maintain a follow-up system to expedite rapid disposition of all change requests. Each time a drawing is changed it is possible to check for incorporation of all approved change requests. See PRODUCT ENGINEERING, March 1946, pp 226-229, for change request and drawing change procedures.

A definite procedure for handling drawing changes is necessary to insure efficient accurate revisions. This should establish methods for: (1) Preparing the drawing alteration; (2) identifying the drawing location of changes; (3) listing all changes requests and advance information incorporated in the revision; (4) identifying the revision by date and letter (or number) in a specified drawing location; (5) detailing the nature, reason for and effectiveness of each item in a notice-of-change distributed with prints of the revised drawing; and (6) checking and approving the changed drawing and its notice-of-change prior to release of revised prints.

Master Parts Lists

A master parts list is required whenever the drawings describe units involving subassemblies and detail parts. This is necessary to guide other departments in ordering materials, fabricating proper quantities of parts, and correctly assembling finished articles. Preparation and issuance of accurate, up-to-date master parts lists is equal in importance to preparing the drawings.

Engineering problems do not end, regretably, with completion of initial release of a set of drawings describing a particular article. Throughout production of the article there will be requests for drawing interpretation, additional information, drawing changes to facilitate production, and advice on the best method of salvaging improperly manufactured parts. This results in a constant flow of requests to, and information from, the engineering department.

Unless these requests are channeled through a specified point there will be needless duplication, and frequent release of erroneous or confusing information. Such a condition is obviously intolerable for, on the one hand it increases engineering expense, and on the other produces inter-departmental discord.

The solution to this problem is simple. A member of the engineering department, designated as liaison engineer, is assigned to receive all requests, obtain the information, and forward the answer to the requestor. Such an arrangement will work well when a competent person is chosen as liaison engineer, and emphasis is placed upon all requests receiving a speedy answer. Rapid handling of requests can be achieved by delegating to the liaison engineer authority to decide a majority of requests without necessity of obtaining executive approval.

All organization, planning, personnel control, methods improvement, and the like, aim toward a common goal, the completion of engineering assignments within established budgets of money and time. The only purpose in carefully organizing and controlling department operations is to insure achievement of this end.

When planning is based upon sound estimates and realistic concept of the time required to accomplish engineering work, maintaining a schedule is largely a matter of daily comparison of actual with scheduled progress, with necessary corrective action applied whenever and wherever work is behind schedule. Illness, termination or training of employees are not acceptable reasons for failure to meet schedule. These conditions are likely to arise during any project. Suitable allowances should have been made. Only major disasters beyond control of the engineering department are valid reasons for failure to meet schedule.

The causes of lagging schedules are few. Insufficient manpower, failure to properly apply available manpower, incorrect original estimates, unnecessary work, or engineering error, excepting the first, are in reality reflections of incompetent leadership. When the cause is ascertained, the corrective measures, which should be carried out, are usually obvious.

ENGINEERING ABSTRACTS

FOREIGN AND DOMESTIC ARTICLES

Plate Spring Design

Foreign Abstract from "The Plate Spring" by K. Walz in V.D.I.-Zeitschrift, Nov. 25, 1944.

THE SIMPLE CALCULATIONS in designing plate springs have been applied since 1903. Nevertheless the use of these springs has been restricted partly because of false rumors of inefficiency, and the necessity of guiding the spring by shells or pins. Therefore, a false impression is prevalent that plate springs are adaptable only to continuous loads. But experience has shown that plate springs afford many advantages when applied to fluctuating and surging loads.

There are several methods of making the requisite calculations to determine the inside diameter d, the outside diameter D, the metal thickness S, and the angle α of the cone as indicated in Fig. 1. Usually the simple procedure of solving equations covering stresses in annular plates suffices for accuracy, providing that the plate angle does not exceed 7 deg., and that the stress σ at the inner circumference does not exceed 180 kg. per sq. mm. The spring load P, in terms of ϕ_1 and ϕ_2 that are dependent upon the diametrical ratio d/D taken from Fig. 2, is

$$P = \frac{S^2 \sigma}{\phi_2}, \text{ kg.} \quad (1)$$

The elastic deformation f of a plate is

$$f = \frac{\phi_1 D^2 \sigma}{4 S}, \text{ mm.} \quad (2)$$

and the plate constant C is

$$C = \frac{P}{f} = \frac{4 S^3}{\phi_1 \phi_2 D^2} \text{ kg. per mm.} \quad (3)$$

Corresponding with the peak of ϕ_1-curve, the maximum efficiency of the plate is attained when the ratio d/D equals 0.475. The ratio S/D should have a minimum value of 0.03 and not exceed 0.06.

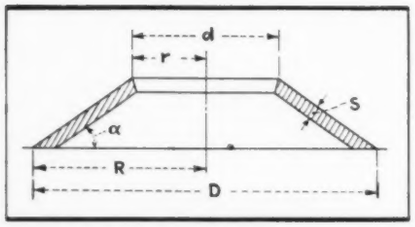


Fig. 1—Spring plate with exaggerated cone angle.

Load-travel curves for three springs are shown in Fig. 3, assembled with ten superimposed plates. These curves are based on experimental observations. Comparison with corresponding calculated results can be made by referring to Table I. For plate (2) the calculated and the experimental values coincide. The other two plates were about 10 percent harder and less accurate, because of the flatter angle α and smaller outside diameter D. A deviation of +10 percent is therefore observed in the results.

The work A done by the spring plate can be derived by combining Equations (1) and (2) and using ϕ_3 for the ratio ϕ_1/ϕ_2, or

$$A = \frac{Pf}{2} = \frac{\phi_3 \sigma_2 S D^2}{8} \quad (4)$$

When the ratio S/D is less than 0.03, the results of radial stress appear as the edge around the inside circumference shows signs of collapse. Therefore plates having a ratio of 0.03 are unstable. When the ratio exceeds 0.06, the calculations for stresses in the plate are no longer reliable.

The volume V of material in one spring plate is

$$V = \pi S \left(\frac{D^2}{4}\right)\left(1 - \frac{d^2}{D^2}\right), \text{ cu.mm.} \quad (5)$$

A factor K that may be regarded as a measure of the utilization of material having a modulus, E, kg. per sq. mm., is defined as

$$K = \frac{E \phi_3}{2 \pi (1 - d^2/D^2)} \quad (6)$$

of by substituting Equations (4) and (5)

$$\frac{K V}{E} = \frac{\phi_3 S D^2}{8} \quad (7)$$

Then by combining Equations (4) and (7) the load absorption of a single plate can be expressed as

$$A = \frac{K V \sigma^2}{E} \quad (8)$$

The factor K will reach a value of 0.5 under favorable conditions when the tensile stress σ is distributed evenly over the cross-section of the element.

The proper utilization of the material from Equation (6) is seen to be dependent on the diametral ratio d/D. Front curve ϕ_3 in Fig. 2, the most efficient loading occurs when d/D equals 0.575. For this condition the factor K has a value of 0.08 and is much worse than for a helical spring of round wire construction where K equals 0.25; and for a spring with rectangular flat members, the thickness of which tapers in accord with a simple parabolic equation, and for which K equals 0.617. From the foregoing it can be noted that the utilization of plate springs is not influenced by a saving in construction material.

Sustained rigidity in its support is characteristic of the plate spring because of the supporting reaction of the bending stress. This advantage makes it feasible for the plate spring to function where helical springs would be impracticable, because of the unfavorable ratio between their average diameter and their wire strength that requires excessive thickness of the spring metal. By superimposing several plate spring units, harmful vibrations can be dampened effectively, a function that is often lacking in helical springs or else

Table I—Calculated and Observed Plate Spring Constants for Different Plate Designs

Plate Design Number	Dimensions				Plate Constants C, kg. per mm.	
	D, mm.	d, mm.	S, mm.	α, deg.	by Calculation	by Experiment
1	21	10	1.2	6½	510	560
2	27	18.5	3.5	5	1,410	1,428
3	36	16.4	2.45	4½	4,170	4,610

Table II—Spring Constants of Plate Springs Superimposed Into Banks of Multiple Tiers

Spring Plate Type	Grouping	Spring Constant *C*, kg. per mm.			
		Number of Plates in each Group			
		1	2	3	4
1	Single group	560	1,130	1,820	2,200
	Ratio of multiple to single tier	1	2.02	3.25	3.95
2	Single group	1,428	3,180	4,680	6,220
	Ratio of multiple to single tier	1	2.24	3.29	4.38
3	Single group	4,610	10,400	14,430	18,900
	Ratio of multiple to single tier	1	2.25	3.13	4.08

would also require several springs to act in series.

Two or more spring plates are often grouped together in a unit as shown in Fig. 4. The relation between load and travel of plate spring columns having a no load height l_o of approximately 148 mm., composed of groups of one, two, three and four of the same spring plate are given in Table II.

These results indicate that for n plates in series to obtain the same deflection it would be necessary to apply approximately n times the load required by a single plate. Deviation to higher ratios would result from friction. In accord with the n times increased force, the load absorbing capacity of the groups also increases n times. Since the same standard plates are used throughout the series, the load stress is evenly divided and does not exceed that of a single plate.

The n times increased load absorption without increasing the plate stresses and without utilizing superimposed groupings is also attainable, as is indicated by Equation (4), if the single plate unit could be made n^3 times as thick. This relation indicates that superimposition of single plate elements for a given load absorption and plate thickness does not result in a reduction of the plate stresses. As already established the superimposition of n plate spring units will increase the constants of the single spring n times.

If the spring plate thickness is increased n times, as may be calculated by Equation (3), the multiple tier banking of the single plate springs therefore makes it possible to reduce the plate constant, without altering the spring loading or increasing its stresses.

The single unit plate spring is more sensitive against distortion in excess of permissible limits than is a group of superimposed plates. The metal thickness of these individual plates is less and consequently they are under lower stress for a given load. Should overloading occur, it has been shown by experience that an individual plate of the group will break. If the column of superimposed plates is surrounded by a shell, broken metal will be prevented from scattering, the column will not be changed in height and can continue to function. It is possible thereby that dampening action may be slightly increased. As shown by Table III the efficiency of the group units decreases when the number of plate units is diminished.

The calculations for the permissible stresses are dependent on the character of the finishing and the composition of the material. For constant loads a maximum remote fiber stress of 300 kg. per sq. mm. is permissible. If the load is a combination of static and dynamic deformations, the maximum fiber stress must be reduced more drastically when these stresses bear a greater share in producing the total deformation. The most favorable condition occurs under load surges during which the stresses fluctuate between zero and the maximum.

The life of a spring plate under a surg-

Table III — Efficiency, in Percent, of Spring Plates When Banked Into Groups

Spring Plate Type	Number of Plates in Group			
	1	2	3	4
1	95	87	78	76
2	98	85	73	85
3	92	81	72	47

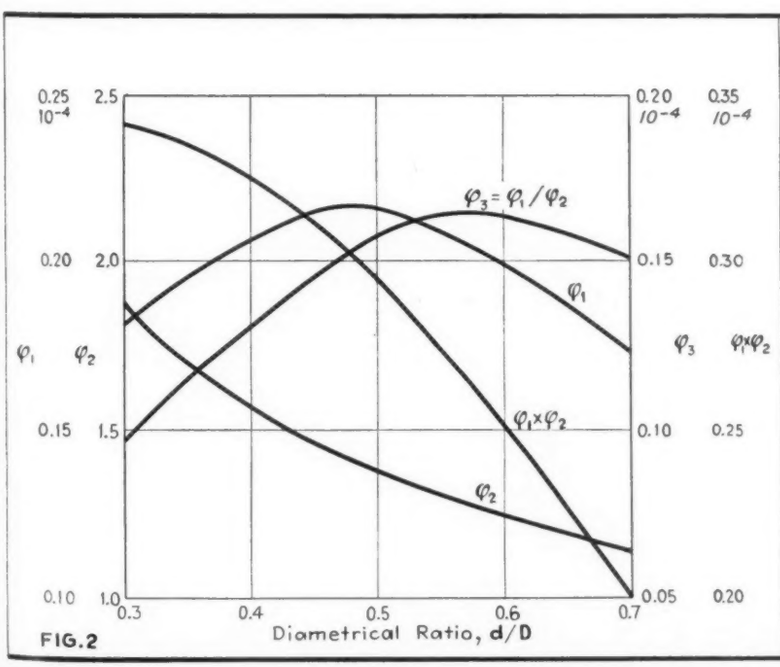


FIG.2

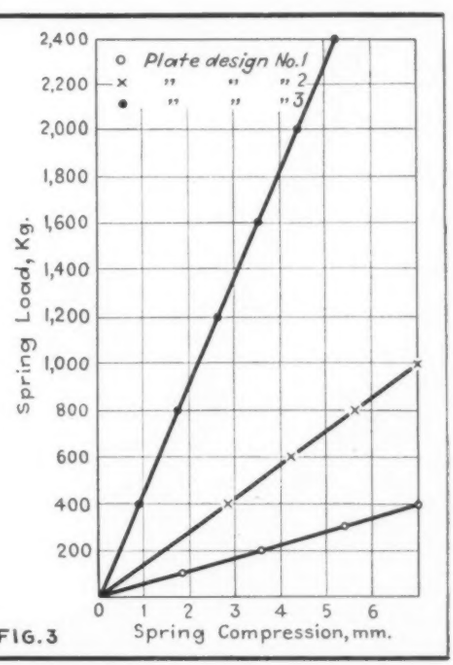


FIG.3

Fig. 2—Factors for use in spring calculations. Fig. 3—Spring load-travel curves for single plate springs having different plate designs.

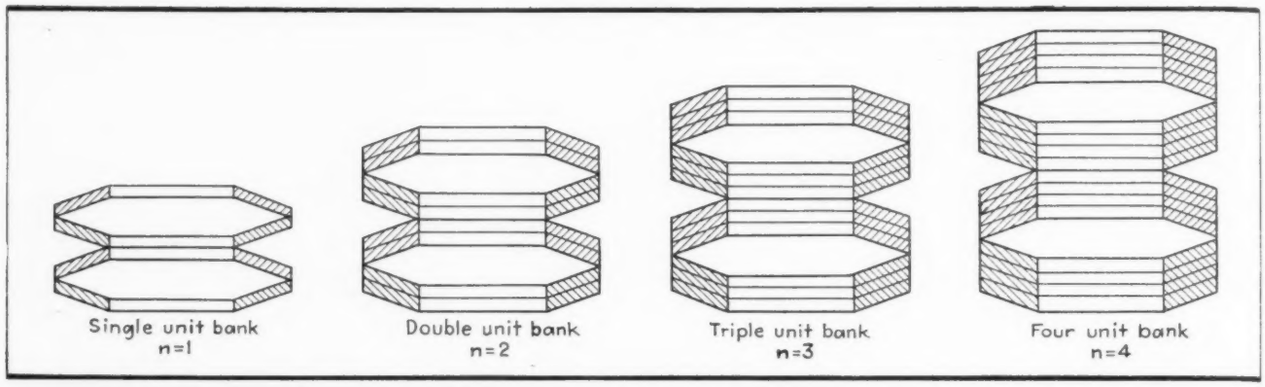


Fig. 4—Plate springs in single and multiple groupings.

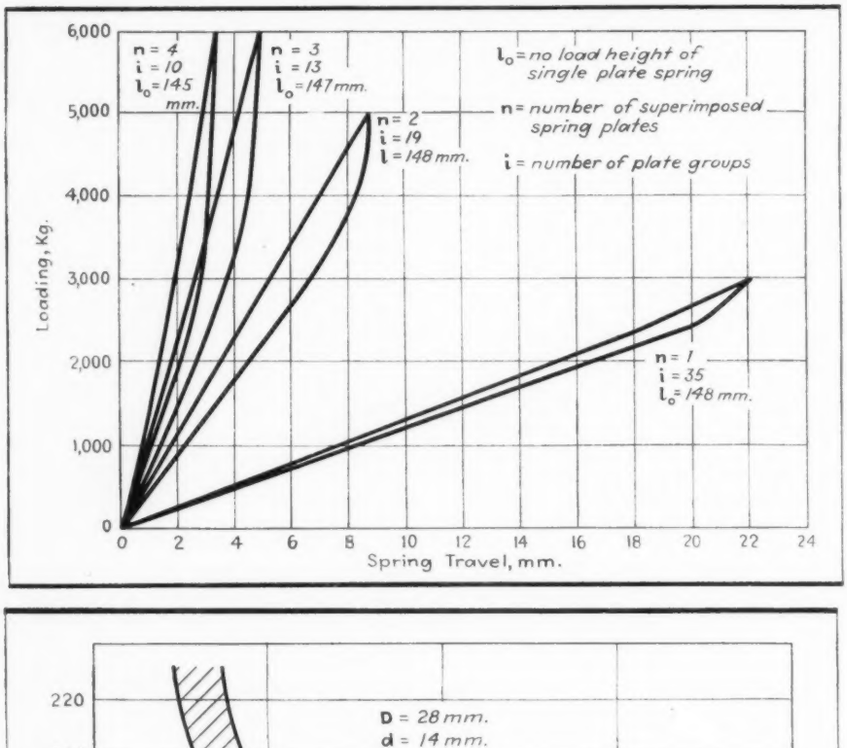


Fig. 5—Force-travel curves for several tiers of plate springs (Type 3, Table I).

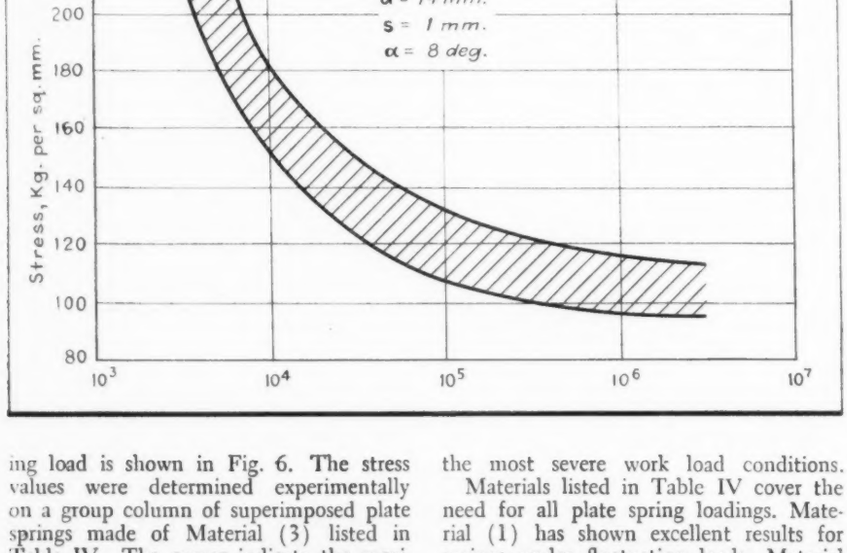


Fig. 6 — Area between Woehler Curves of a spring plate, made of Material (3), Table IV, under fluctuating load.

ing load is shown in Fig. 6. The stress values were determined experimentally on a group column of superimposed plate springs made of Material (3) listed in Table IV. The curves indicate the maximum stress to which the plate springs may be subjected for a given life under the most severe work load conditions.

Materials listed in Table IV cover the need for all plate spring loadings. Material (1) has shown excellent results for springs under fluctuating loads. Material (2) should be used for springs where rigid support and long life are needed. Because this steel has a high silicon content and may become brittle around the edges, it should never be used for fluctuating stress. Material (3) is suitable for severe and intermediate fluctuating stress.

For substitute materials, either hot-rolled steel or cold-rolled steel strip are suggested. The latter is preferable because of its characteristic fibrous structure, whereas hot-rolled steel may vary considerably in thickness. This fault can be corrected by grinding the inside surface of the fabricated spring plate. For plates that are to be subjected to fluctuating loads and that have a low safety factor, faulty fibrous structure can be avoided by rolling the steel equally in two directions at right angles to each other and by wet grinding both sides.

The spring plate disks are stamped oversize and are machined or ground to the finished size, after they are formed. In this process the plate opening is first stamped out for later finishing of its edge. This procedure is unnecessary for plates that are to be subject only to static load. The rounding out of the opening on both sides of the plate is required only when the load is fluctuating.

The plates as they are taken out of the salt bath are hardened in oil or better still in a warm bath at approximately 180 deg. C. The salt prevents harmful de-carbonization and burning of the edge metal. The initial tempering temperature should be chosen to impart a Rockwell C hardness ranging between 40 and 50. This range is to be exceeded only for spring plates that are to be continuously subjected to fluctuating load.

Shot-finishing is applied only to spring plates that will be subjected to fluctuating load. Plates subjected to static load are not treated in this manner because

Table IV—Steel for Plate Springs

Material	C	Si	Composition, percent Mn	P	S
1	0.5 to 0.7	0.45	0.5 to 0.8	0.035	0.035
2	0.55 to 0.7	1.6 to 1.9	0.6 to 0.8	0.035	0.035
3	0.55 to 0.65	0.9 to 1.1	1.0 to 1.2	0.03	0.030

the shot roughens the surface thereby decreasing its resistance to corrosion. The shot cleaned surfaces should receive a rust resistant protective coating, coatings of oil or grease generally suffice. Phosphating increases the adhesiveness of protective coatings without affecting the shot finish. Galvanized zinc coatings likewise have proved to be effective and readily respond to bending. If an acid electrolyte is used the life of the coating is not greatly shortened under fluctuating load. Hard chrome coatings have a limited life, because of the chemical effect of the hydrogen released during plating. Chrome finishes are brittle and may flake.

Before placing the spring plates in service they are overloaded to produce favorable internal stresses.

The assembled plate spring column travel is guided by a surface hardened bolt. If close limits are specified for the unloaded height, the column is provided with an adjustable compensating plate. The correction afforded thereby, makes it feasible to avoid impractical tolerances.

Thermoplastic Coatings for Cloth

From "Plastics as Coatings", by Paul F. Corbin, Textileather Corporation, presented at the annual meeting of the Society of Plastics Engineers, held in Detroit on January 9, 1946.

A COATED FABRIC is a fabric treated with oil, oil and resin, vinyl resin, rubber or synthetic rubber, nitrocellulose or other cellulose esters, or other synthetic resins, in which a coating is primarily applied to the surface, producing a continuous film.

There are four methods of applying coatings to cloth: Spreader or doctor knife coating, calendering, dipping, and rolling. Mr. Corbin's talk was confined to the doctor knife and calendering methods.

There are six general types of coatings as applied to cloth: Cellulose ester, such as nitrocellulose, ethyl cellulose, cellulose acetate, cellulose acetobutyrate; second, vinyl resin coatings, such as the copolymers of vinyl acetate and vinyl chloride, polyvinyl butyral and polyvinyl alcohol; third, oil coating, principally linseed, chinawood, dehydrated castor and fish oil; fourth, oil-in-resin coatings, such as varnish cambric and the coating for laminated plastics, using phenolic and modified phenolic resins

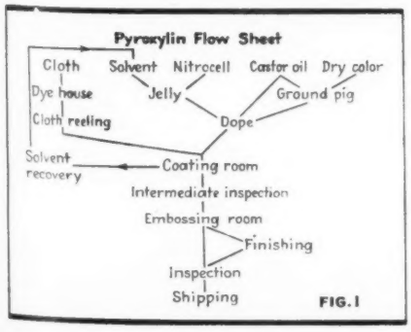


FIG. 1

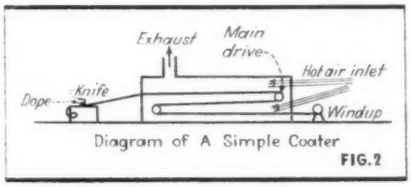


Diagram of A Simple Coater
FIG. 2

in oil; fifth, starch and casein; sixth, rubber and synthetic rubber coatings. The discussion was limited to nitrocellulose and copolymers of vinyl resins.

All coatings generally start with a base fabric, usually cotton, with occasional use of rayon, nylon and fiberglas. The cloth is usually dyed to match the face coating.

Statistics from the Bureau of Census on the amount of nitrocellulose coated fabrics produced in relatively recent years: In 1940, 60 million yards; in 1941, 88 million yards; in 1942, 75 million yards. This yardage was consumed largely in the following fields: Bookbinding; shoe industry; automotive seat coverings; trim and upholstery, metal furniture; hassocks; theater seats; pocketbooks and belts; card tables and chairs; restaurant and cocktail bars; general furniture requirements; luggage and trucks; optical cases and case coverings.

Proxylin coating is a mixture of nitrocellulose, castor oil and pigment ground in castor oil with small amounts of chemical plasticizers for specific properties. The nitrated cotton varies from five- or six-second to thirty-second viscosity with a preference of fifteen- to twenty-second.

Castor oil is generally used as the plasticizer. Chemical plasticizers are dibutyl phthalate, tricresyl phosphate, rincinoleotes, the plasticizing alkyds and sebacates in small quantities, to obtain specific results.

In the organic field of pigments, particularly reds, Scarlet 2R is the most common type. Inorganic pigments used are chrome, the molybdate oranges and cadmium reds. Pigments used in pyroxylin coated fabrics should be stable to light, have good coverage and opacity, be non-crocking, have low oil absorption and must not "bleed" in solvent or castor oil.

A flow sheet for making a proxylin coating fabric is shown in Fig. 1. The "jelly" is made by dissolving nitrated cotton in solvents such as ethyl acetate, acetone, alcohol, methylethylketone and naphtha. Solid content of this jelly varies from 16 to 24 percent, depending on the type of cotton and the viscosity. The pigments are ground in castor oil, usually taking into consideration oil absorption; an arbitrary 60 percent pigment and 40 percent oil ratio is sometimes used.

A plasticizer or oil to give required pliability, is added. These three ingredients, when mixed, form the coating dope. While the coating dope is being prepared, the base textile or cloth is usually dyed with direct dyes, which are water soluble.

From the dye jigs, where the cloth is colored, it proceeds to a stack of drying cans and from there to a tenter frame, which dries the cloth under tension and maintains the width.

The dope is spread on the cloth by means of a doctor knife, and then run into a heated chamber at approximately 200 deg. F. The speed at which this material is run varies from twelve to fifty yards a minute, depending on the weight of the coating applied.

A rough diagram of a conventional coater is shown in Fig. 2. The dope starts at the far end and is spread on by a doctor knife, through a heated box; from one to three passes are required. Sometimes the cloth is returned on an endless conveyor to the front end, necessitating no sewing or seams.

From the coating room the cloth goes to an intermediate inspection room, where major defects are cut out and rolls are prepared for roller or plate embossing and refinishing.

The roller type of embossing is more adaptable for volume production of a single grain, whereas, plate embossing lends itself to diversification. In roller embossing the temperature used is from 210 deg. F. to 225 deg. F. Pressure on journals is 20 to 30 tons. In plate embossing the speed is eight impressions per minute, three or four impressions a minute for vinyl resin coaters.

Characteristics obtainable from vinyl resin coatings, particularly the high chloride content copolymer, are: Chemically resistant fabric; high flex-and-fold-resistant fabrics; non-spewing at 180 deg. F.; free from cold cracking from minus 40 deg. to minus 60 deg. F. high abrasion resistance; resistance to alkali and acids; resistance to oil and solvents; resistance to fire; and resistance to weather.

There are two general methods of applying resin coatings: First, a solution method similar to the pyroxylin coating system just described. The resin used in this system is a soluble type 87 percent

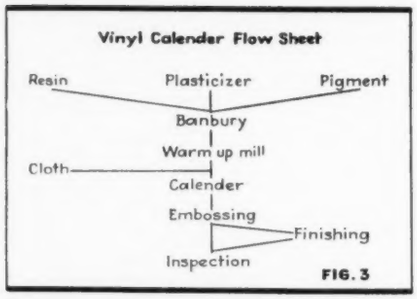


FIG. 3

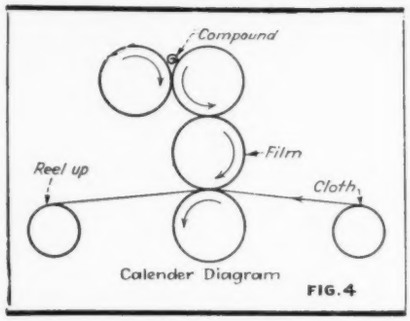


FIG. 4

chlorine copolymer, dissolved in methyl-ethyl ketone. The solids on this type of cloth are relatively low, 25 to 30 percent, and many layers are required to build up to coating weight.

The second method is calendering or applying a plastic film without solvent in one application to a piece of cloth. The choice of plasticizers, which are numerous, has great bearing on the final product, as each plasticizer or combination of plasticizers imparts special properties to the finished film. Pigments also must be chosen with great care to prevent reaction with the resin itself or the stabilizer. There must be no free sulfur present. Pigments must stand up at 350 deg. F. in order to be stable to milling and calendering.

A flow sheet of material through a vinyl resin calendering plant is shown in Fig. 3. Resin, plasticizer and pigment are mixed in a Banbury, and put through a warm-up mill. At the calender the cloth meets the resin; it runs from the cold calender, is embossed and goes to inspection.

After the cloth is calendered, it is then embossed in a process similar to proxylin embossing, but at different temperatures and speeds than in the proxylin system. It has been found that a rather high temperature preheating is necessary before embossing the vinyl resin coating. Finally, the cloth is given a finish or special effect, and sent to the final inspection.

The modern resin calender usually has four rolls (Fig. 4). The top two rolls receive the charge from the warm-up mill, and are in a horizontal plane. The second, third and fourth rolls are in a vertical plane. The film is formed between rolls one and two, and is cast on the third roll and applied to the cloth between the third and fourth rolls.

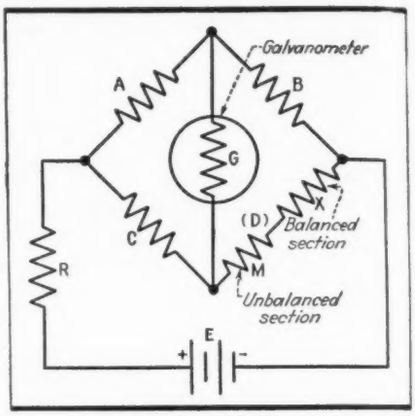


Fig. 1—Unknown arm, D, is divided into balanced and unbalanced sections.

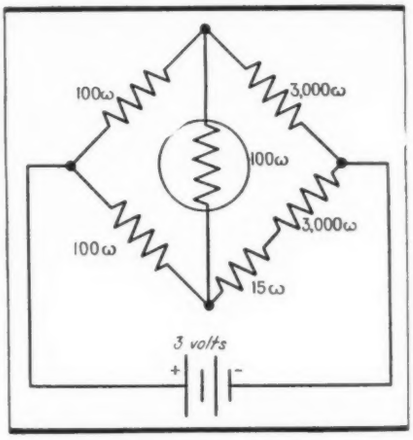


Fig. 2—Typical analysis showing 15 ohms as the unbalanced section in one arm.

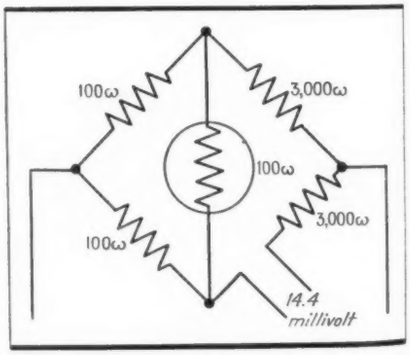


Fig. 3—Unbalanced resistor replaced by an injected voltage.

Simplified Bridge Analysis

From "Simplified Bridge Analysis" by John H. Miller, chief engineer, in Weston Engineering Notes. Volume 1, No. 3, June 1946, published by Weston Electrical Instrument Corp., Newark, N. J.

To DETERMINE the galvanometer current within a few percent in a Wheatstone bridge, the following simplified method of analysis quickly reveals the order of the sensitivity required in the galvanometer for a given degree of unbalance. This approximate method is especially helpful in selecting galvanometers from an instrument-room or a catalog. While not strictly rigorous, particularly for large unbalanced conditions, its accuracy does increase as the balance point is approached.

Referring to Fig. 1, the four arms of the bridge are conventional except that the fourth arm, *D*, consists of two parts, *X* and *M*, where *X* is the value that would balance the bridge and *M* is the difference between this and the actual value.

Since it is the voltage drop of the current passing through *M*, the unbalanced portion, that actuates the galvanometer through the bridge network, the evaluation of this voltage drop and its application into the galvanometer and its associated network will give the resultant galvanometer current.

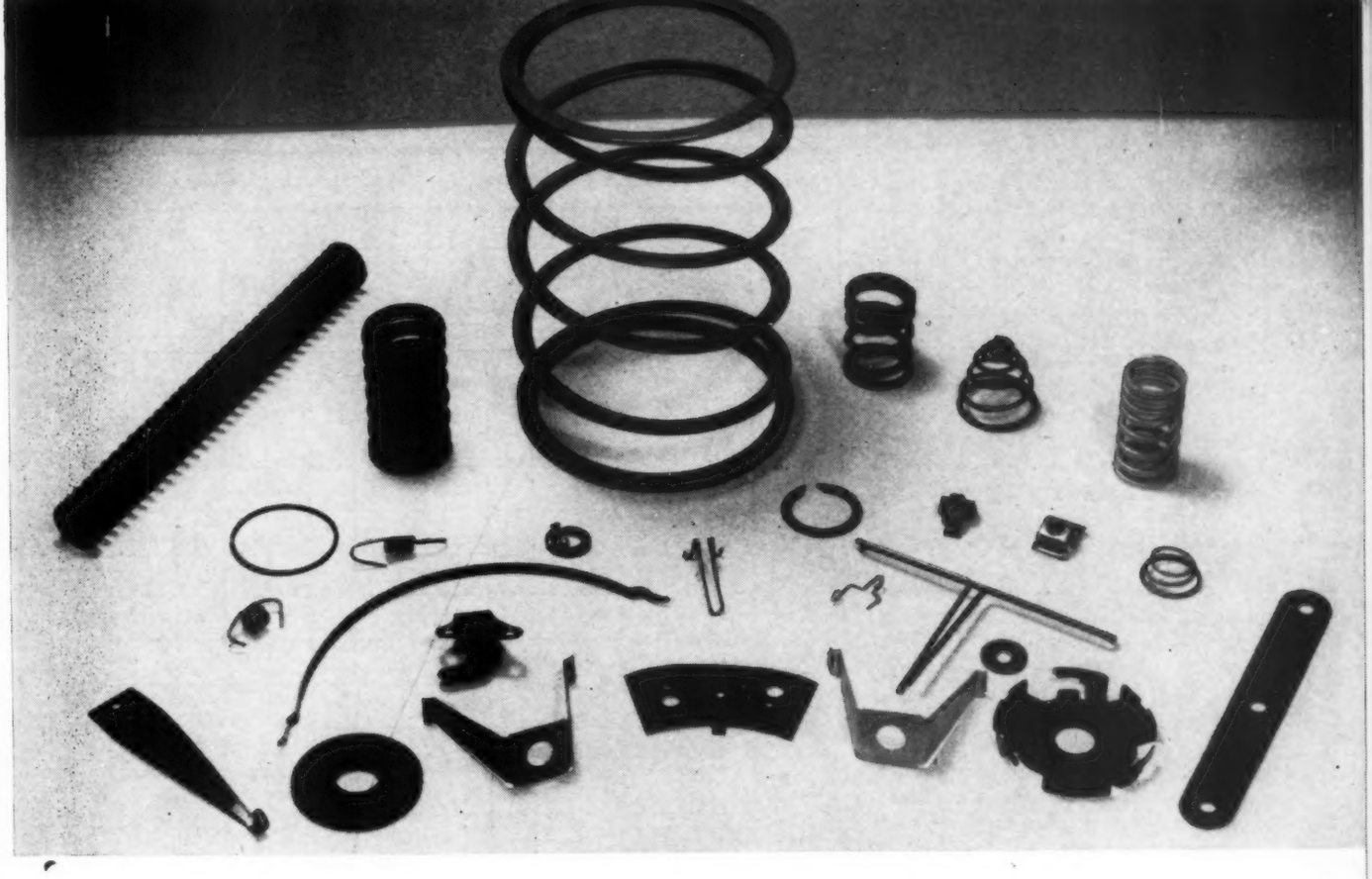
In the ordinary bridge application the resistance, *R*, is usually negligible. Since the bridge is close to balance, the current through the galvanometer can be neglected in obtaining the current through *C* and *D* (or *C*, plus *M* plus *X*) so these two arms are in series across the battery potential *E*. For example, in determining the galvanometer current for the typical values in Fig. 2 a resistor with 3,015 ohms is measured with the bridge actually balanced for 3,000 ohms. With a 3 volt battery and a resistance in the two lower arms of 3,115 ohms, approximately 960 microamperes flows. This 0.96 milliampere current flowing through the 15 ohm resistor produces a 14.4 millivolt drop.

As shown in Fig. 3, the next step is to discard the battery and inject this value of 14.4 millivolts into the network in place of the 15 ohm unbalanced component, and again evaluate the result. The circuit has 6,000 ohms in the right hand arms plus the resistance of the galvanometer shunted by the left hand arms. The two parallel circuits of 100 ohms and 200 ohms are the equivalent of 67 ohms, so the total is 6,067 ohms. With the 14.4 millivolts applied to 6,067 ohms, by slide rule, approximately 2.38 microamperes will flow.

Of this current, 2/3 will flow through the galvanometer since its resistance is 100 ohms shunted by 200 ohms. By slide rule, 2/3 of 2.38 gives about 1.57 microamperes as the solution.

Solving the same problem by rigorous equations, the exact value is 1.587 microamperes.

If the resistance under test is less than the balanced value, a resistance is used in a negative sense to secure balance. In the example given, the same values would be used if arm *D* were 2,985 ohms, except the 15 ohm value would be preceded by a negative sign. The numerical results are the same, but the galvanometer would deflect in the opposite direction.



Carbon and Alloy Steel Materials For Cold-Formed Springs

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Chief Engineer, Barnes-Gibson-Raymond Division,
Associated Spring Corporation

Properties of the types of steel commonly used for the manufacture of cold-formed springs. Special emphasis is given to the proper choice of material to meet various service conditions such as impact loading, elevated temperatures or corrosive atmospheres.

SPRINGS used in engineering applications can, in general, be classified either as hot-formed or cold-formed springs depending upon the method of fabrication used in their manufacture. The choice of fabrication method, of course, is dictated primarily by the size of the stock and the shape to which the spring is to be made.

Cold-formed springs are made in flat shapes in thicknesses ranging from 0.001 to 0.100 in. and in widths usually less than 3 in., and in round shapes having diameters from 0.002 to 3/8 in. Square and rectangular shapes are usually limited to the same total cross-sectional area as the round springs. Most cold-formed springs are formed on high speed, automatic machinery designed for volume production at minimum cost.

From the variety of spring materials available in both ferrous and nonferrous metals, it is possible to choose for any given spring application, a material that will have the necessary physical or chemical characteristics and, at the same time, present maximum economies in material cost. Thus, there are available cold-drawn or oil tempered straight carbon steels of different qualities with tensile strengths ranging from 200,000 to 500,000 lb. per sq. in.; stainless steels for corrosion resistance; nonferrous materials for electrical conductivity; and high speed tool steel analyses for the fabrication of springs to operate at high temperatures.

Carbon Steels for Flat Springs

Stock for the fabrication of flat steel springs is readily available in three plain carbon steels, the analyses of which are given in Table I. These steels can be obtained in either the hardened and tempered or the annealed condition. The S.A.E. 1095 steel should always be used in the pretempered condition.

Ordinary clock springs of S.A.E. 1095 steel are stressed to 200,000 lb. per sq. in. in thicknesses of 1/16 in. while springs 0.110

in. thick are stressed to 250,000 to 275,000 lb. per sq. in. A cantilever spring of finite life, a stress of 180,000 lb. per sq. in. can be employed. If a spring must give infinite life, the endurance range varies with thickness. When the springs are to be subjected to heat, stresses must be lowered as the temperature increases, until above 400 deg. F. the permissible stress becomes so low as to render useless springs made from S.A.E. 1095 steel.

The three normal hardness ranges, in which this steel is available to commercial users, are Rockwell C 51-C 48, C 47-C 45, and C 45-C 41. The material has a modulus of elasticity of 30,000,000 lb. per sq. in. The tensile strength varies with the hardness and thickness from 310,000 lb. per sq. in. at 0.005 in. thick with the highest hardness to 190,000 lb. per sq. in. at 0.065 in. thick with the lowest hardness.

The two A.I.S.I. steels C1074 and C1064 can be obtained either pretempered to the same Rockwell ranges as S.A.E. 1095 steel or in the annealed state. When used in the hardened state, these steels form better at equal hardnesses as their carbon and manganese contents approach the lower portion of the analysis range. They are used in the fabrication of springs with contours too complicated to permit the use of pretempered S.A.E. 1095 steel.

When exceptionally difficult forming is encountered, the A.I.S.I. steels in the annealed condition should be used. In the subsequent heat-treatment, more uniform results are usually obtained from the A.I.S.I. C1074 material. The drawing temperature is higher than that for the 0.60 to 0.70 percent carbon steel and the resulting agreement with specified dimensions is closer when the springs are set in fixtures. Springs more than 1 in. wide and 1/16 in. thick should only be made of the 0.70 to 0.80 carbon steel. At temperatures higher than 400 deg. F., these steels cannot be used at any stress without load loss. In Fig. 1 are given the fatigue resistance of A.I.S.I. C1074 steel in various thicknesses.

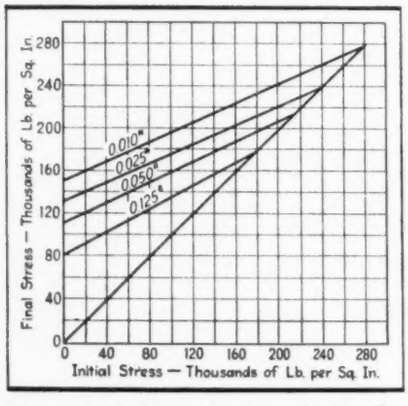


Fig. 1 — Safe endurance limits for 0.70-0.80 carbon flat steel springs, 45-48C Rockwell, tested as cantilevers.

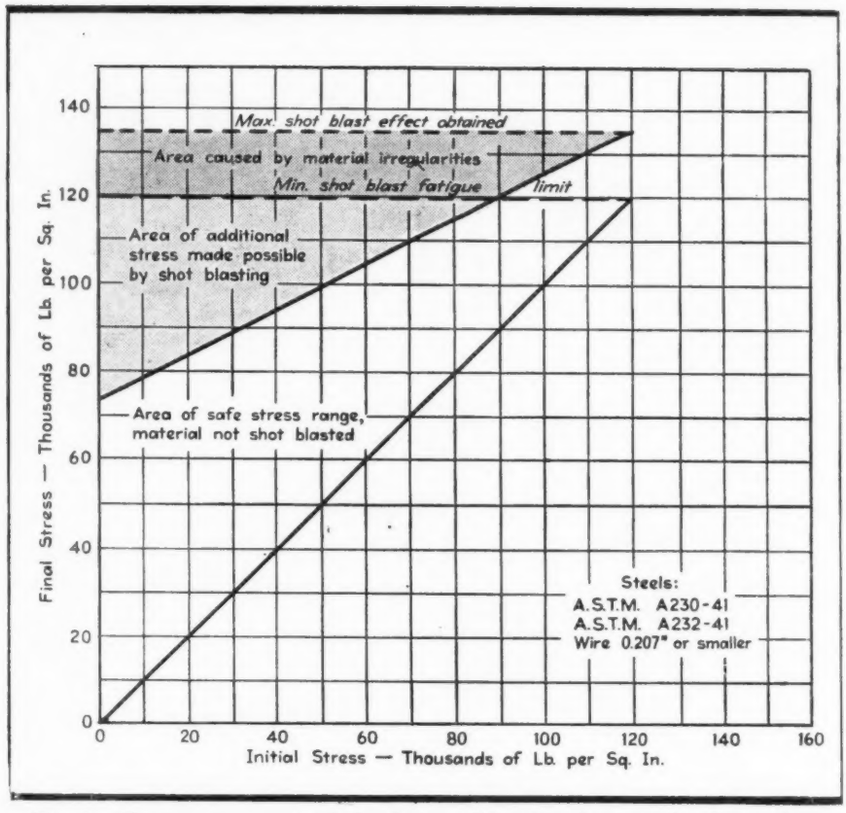


Fig. 2—The improvement of the fatigue life of carbon steel valve springs caused by shot peening. Stress corrected by Wahl Factor.

In an endeavor to avoid the added cost of carbon extras, some engineers specify a 0.40 to 0.50 percent carbon steel for thin sections. This steel will make some forms of clips, and parts that are subjected to static load, when the sections are thin and narrow, but the practic saves little in the long run. At times steel of poor hardenability is received and the resulting loss in defective parts will offset the saving. When used on sections thicker than 0.035 in., soft spots are sure to occur.

Flat steel in alloy grades, such as S.A.E. 6150, or 9260, are infrequently used since in thin sections the higher hardenability is not needed and the fatigue life does not exceed that of carbon steel. These steels are sometimes used to resist heat at higher stresses than carbon steel will stand.

The edges of all flat springs should be nicely smoothed and rounded, if they are expected to withstand many flexures. One of the major causes of failure in highly stressed flat springs is the neglect to polish sheared edges. The omission of this operation leaves scratches and roughness that raise stresses to the failure point.

Low Stressed Springs

A major portion of the total output of cold-formed springs is destined for use at low stress levels in applications such as upholstery springs and actuating springs for mechanical gadgets. Because of the high tonnages of steel required, every effort is made to use as inexpensive a material as possible. Maximum economy is achieved through the use of plain carbon steels that are cold-drawn to the required diameter and formed into springs without further heat-treatment except that incidental to the baking of enamel coatings.

The steel used generally contains 0.40 to 0.70 carbon and 0.80 to 1.20 manganese. The cold-drawing operations produce tensile strengths in the range of 200,000 to 300,000 lb. per sq. in. Since neither the chemical analysis nor the physical properties of this type of steel are carefully controlled, it is not advisable to use it for any important application.

Highly Stressed Cold-Drawn Wire Springs

The best quality cold-drawn wire, known as music wire, is made from plain carbon steel containing approximately 0.80 carbon and 0.60 manganese. Between several of the drawing operations, the music wire stock is patented (heated above its critical temperature by passing it through a lead pot and then quenching in lead) to produce a uniform sorbitic structure. The last drawing operations are carried out under a liquid containing tin salts. These salts produce a slight deposit of tin on the

wire that lubricates it during passage through the die and results in a smooth shiny surface. The total reduction of cross-sectional area is approximately 80 percent.

This careful fabrication procedure produces a wire of great toughness, and uniformly high tensile and torsional strengths. If heated to 375 to 425 deg. F. after forming into springs, the hardness, and tensile strengths will increase. Heating to 700 to 725 deg. F. will cause a slight drop in physical properties but will increase the fatigue life and the resistance to heat.

For springs that are formed of wire less than 0.028 in. in diameter, music wire is generally used since the high cost of drawing wire in smaller sizes does not permit much reduction of cost through the use of inferior materials. Music wire can be used in sizes up to 1/8 in. dia. when its surface finish or high physical properties are necessary. In sizes over 1/8 in. dia., it is generally more economical to use a heat-treated wire since it will possess equivalent physical properties because the properties of music wire decrease with increased wire diameter.

Oil-Tempered Wire For Mechanical Springs

The majority of mechanical spring applications, in which any degree of dependability and uniformity are required, are fabricated from oil-tempered wire. This wire is usually made of plain carbon steel containing 0.60-0.70 carbon and 0.60 to 0.90 manganese and is patented and cold-drawn to size. In drawing oil-tempered stock to wire size, the reduction of cross-sectional area is only 50 to 70 percent.

After having been drawn to its final diameter, the wire is heated to 1475 deg. F., oil quenched and tempered to produce desired physical properties. The heat-treatment consists of passing the wire through a heating medium, for example, a lead pot 20 to 25 ft. long, then into an oil quench, from whence the wire enters a drawing medium, usually another lead pot. It is obvious that the temperatures and wire speeds must be adjusted so that the wire will be well over the critical range when quenched, and will be drawn to the correct tensile and hardness values in the tempering bath. In leaving the quenching oil, the wire must be hot to avoid the possibility of cracking. For years this process has been martempering steel long before the term was used or the value of the process demonstrated.

By proper regulation of the heat-treating cycle, oil-tempered wire can be produced with tensile strength ranging between 155,000 and 300,000 lb. per sq. in. It has a modulus of elasticity in bending of 30,000,000 and in torsion of 11,500,000 lb. per sq. in. The hardness usually ranges from Rockwell C 42 to 46.

A high quality oil-tempered carbon steel wire, known as valve spring wire, is produced for use in applications in which springs of maximum dependability, fatigue life and uniformity of strength are required. Although its chemical analysis is similar to that of the regular oil-tempered wire, its tensile properties are much more uniform and it is drawn much more carefully than ordinary oil-tempered wire. The heat-treatment is carefully controlled to insure uniform carbide solution and every precaution is taken to avoid surface scratches and decarburization.

Because of its good surface condition and controlled heat-treatment, valve spring wire is exceptionally uniform in load carrying characteristics and possesses excellent fatigue properties. Most valve springs are given a shot peening treatment to produce the best possible fatigue characteristics. Fig. 2 shows the improvements brought about by this treatment.

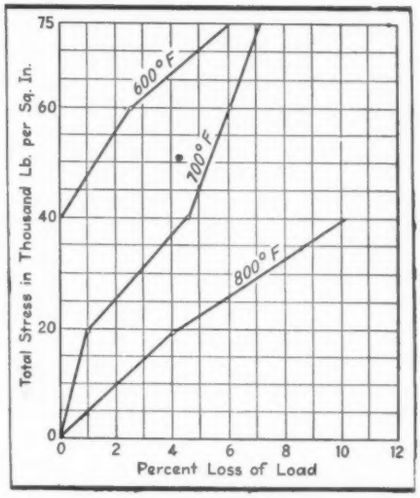


Fig. 3—Load loss characteristics of 18-4-1 high speed steel springs at elevated temperatures.

Alloy Steel Wires For Special Service Conditions

Several types of alloy steel wire are produced for spring applications in which resistance to load loss at elevated temperatures or impact resistance are necessary. The most commonly used alloy steel wires are chrome vanadium, S.A.E. 6150; silico-manganese, S.A.E. 9260; and a new chrome silicon steel that has no specification number as yet. Table II gives the analyses of these steels.

These steels are processed into wire and heat-treated in the same manner as the valve spring grade of oil-tempered wire. They are usually heat-treated to the same physical property range as valve spring steel, although they can be procured at a hardness of Rockwell C 50-53 for use under extreme stresses.

In cold-formed springs, the wire diameters of which are usually less than 3/8 in., the hardenability of plain carbon steel is sufficient to permit through hardening, so that the use of the greater hardenability of the alloy steels does not offer any increase in tensile or fatigue properties. However, both the impact resistance and resistance to load losses at elevated temperature of the alloy steel springs are definitely better than those of the plain carbon type.

When service conditions require either good impact resistance or the ability to resist loss of load at temperatures over 375 deg. F., the use of alloy spring steels is indicated. The chrome vanadium steel is somewhat inferior to the chrome silicon and silico-manganese steels in both impact and temperature resistance. But since the

Table I—Carbon Steels for Flat Springs

Element	Steel Type		
	S.A.E. 1905	A.I.S.I.–C1074	A.I.S.I.–C1064
C	0.90–1.05	0.70–0.80	0.60–0.70
Mn	0.40–0.50	0.50–0.70	0.50–0.70
P max.	0.040	0.040	0.040
S max.	0.050	0.050	0.050

Table II—Alloy Steels for Cold-Formed Springs

Element	Steel Type		
	S.A.E. 6150	Cr–Si	S.A.E. 9260
Carbon	0.48–0.55	0.50–0.60	0.55–0.65
Manganese	0.65–0.90	0.60–0.80	0.70–0.90
Phosphorus (max.)	0.040	0.040	0.040
Sulphur (max.)	0.040	0.040	0.040
Chromium	0.80–1.10	0.50–0.80	.....
Silicon	0.20–0.35	1.30–1.60	1.80–2.20
Vanadium	0.15 min.	.....	.....

chrome vanadium steel is much more easily drawn than the other alloys, it is usually more readily available and is suitable for most applications.

When springs must withstand elevated temperatures, the best results are obtained with steels having the highest carbide concentration. Thus, 18-4-1-tungsten high speed steel or the 6-6-2 molybdenum type are the most suitable steels for high temperature service. At a total combined stress of 40,000 lb. per sq. in., temperature of 700 deg. F. can be withstood. At a stress of 20,000 lb. per sq. in., springs that are exposed to temperatures of 800 deg. F. have been produced successfully. Fig. 3 shows the load sustaining characteristics of 18-4-1 high speed steel at temperatures up to 800 deg. F. In the heat-treatment of these alloys, the use of a double draw to insure relief of any martensitic precipitation is essential. Usually a draw of 1150 deg. F. and cooling to room temperature followed by a second draw at 1250 deg. F. will give the best results.

For use where a spring is exposed to corrosive action, several types of stainless steels have been used with varving success, the best results having been obtained with the 18-8 analysis. This material has been produced in sizes from 0.003 up to 5/16 in. dia. with good results. Larger wire for mechanical springs has not been demanded in any volume. The tensile strength ranges from a minimum of 350,000 lb. per sq. in. for the small size to 160,000 lb. per sq. in. for the large size. Except when grounded or in electrical contact with copper alloys in the presence of halides, this material is highly resistant to corrosion. Its stress range is about 20 percent less than valve spring wire for sizes of 5/32 in. or less and 1/3 less as sizes increase to 5/16 in. The steel shows good resistance to heat and if properly treated and pre-set, after forming into springs, the material will set less than steels other than high speed steel at 500 deg. F. The amount of set at 600 deg. F. sometimes precludes its use for accurate results.

One difficulty encountered in using 18-8 steel is that its modulus varies with different wire mill practices. An adequate stress relief after forming does much to correct this fault. Following a proper relief the steel approaches a modulus of 28,000,000 in bending or 10,000,000 in torsion. Stress relief should not be less than 825 deg. F. A temperature of 925 deg. F. will not harm the physical properties. Stainless steel is not nonmagnetic despite its being listed as an austenitic steel. The smaller the wire or the thinner the flat sections, the more likely it is to be magnetic, as the work of drawing or rolling gives rise to martensitic particles.

Stainless steel of the 16-2 type has the same modulus as steel. Its use has been somewhat limited because the price differential is not great, and with 18-8 available, the demand for its development is light. The use of 18-8 stainless steel with molybdenum additions of 2 or 3 percent and of A.I.S.I. type 36, have been successful. The moly-stainless was much used by the Navy during the war because of its improved corrosion resistance in salt water and reduced electrolytic action. The molybdenum addition causes stainless steel to harden more readily when cold-worked, so that the tensile strength cannot be as high as that of 18-8, if sharp bends or small ratios of wire size to spring diameters are encountered. Lower stresses are advisable because of these conditions.

A heat-treatable stainless steel of the 18-8 type called Stainless W has recently been announced. Its analysis is:

Carbon	0.12 max.
Manganese	2.00 max.
Phosphorus	0.04 max.
Sulphur	0.04 max.
Nickel	6.00–8.00
Chromium	16.00–18.00
Titanium	1.00 max.
Aluminum	1.00 max.

The material is furnished in the annealed condition and is capable of developing the mechanical properties as outlined in Table III. The steel is magnetic. At present no fatigue values are available. While not a true high temperature steel, this material possesses some desirable properties up to 800 deg. F., as shown in Table IV. In this steel the most important element is titanium, which acts as both the primary precipitation hardening element and a strong ferrite former. With all other elements properly balanced, the optimum percentage of titanium has been found to be in the range of 0.40 to 1.00 percent.

Table III—Mechanical Properties of Stainless W

Condition	Tensile Strength, lb. per sq. in.	Yield Strength, at 0.2 percent offset	Hardness, Rockwell C
Annealed	120,000 to 150,000	75,000 to 115,000	22 to 28
Aged at 950 deg. F	195,000 to 225,000	180,000 to 210,000	39 to 47
Aged at 1,000 deg. F	190,000 to 220,100	170,000 to 210,000	38 to 46
Aged at 1,050 deg. F	170,000 to 210,000	150,000 to 185,000	35 to 43

Table IV—Short Time Tensile Strength of Stainless W

Temp., deg. F.	Yield Strength at 0.2 percent offset, lb. per sq. in.	Ultimate Tensile Strength, lb. per sq. in.	Elongation in 3/4 in., percent	Reduction of Area, percent	Modulus of Elasticity, 10^6 lb. per sq. in.
60	182,000	193,000	12.0	52.0	29
600*	156,000	162,000	12.0	51.5	....
800†	135,000	145,000	13.5	53.0	27
1,000	54,000	94,100	22.5	75.0	20
1,200	24,000	41,000	40.0	88.0	14

* Stress to produce 0.2 percent permanent deformation in 500 hr. is 125,000 lb. per sq. in.
† Stress to produce 0.2 percent permanent deformation in 500 hr. is 40,000 lb. per sq. in.

Heat-Treatment Of Cold-Formed Springs

In considering the fatigue life of a spring, which because of its dimensions must be made of annealed material, the method of hardening and tempering should be well worked out for the part. A comparison of the fatigue life of (1) the ordinary quench in oil and drawing back to the desired hardness, (2) austempering to the same hardness, and (3) martempering by quenching in salt at 425 deg. F. and then drawing to the same hardness, has shown that:

1. Austempered springs have lower fatigue life than springs made by either of the other methods.

2. A martempered spring and one made by the conventional quench and draw have the same fatigue life within the accuracy of the testing machine.

3. A martempered spring goes out of shape less in heat-treatment than one treated by the usual quench and draw process. An austempered spring changes shape still less so that it might be considered as a heat-treatment, if a lowered fatigue limit is of secondary importance to warpage. Warpage can be controlled in any of these processes by the use of quenching or setting fixtures, but these are expensive and slow the operation.

4. The reduction of area and ductility of austempered or martempered spring steel are greater than those resulting from the quench and draw method.

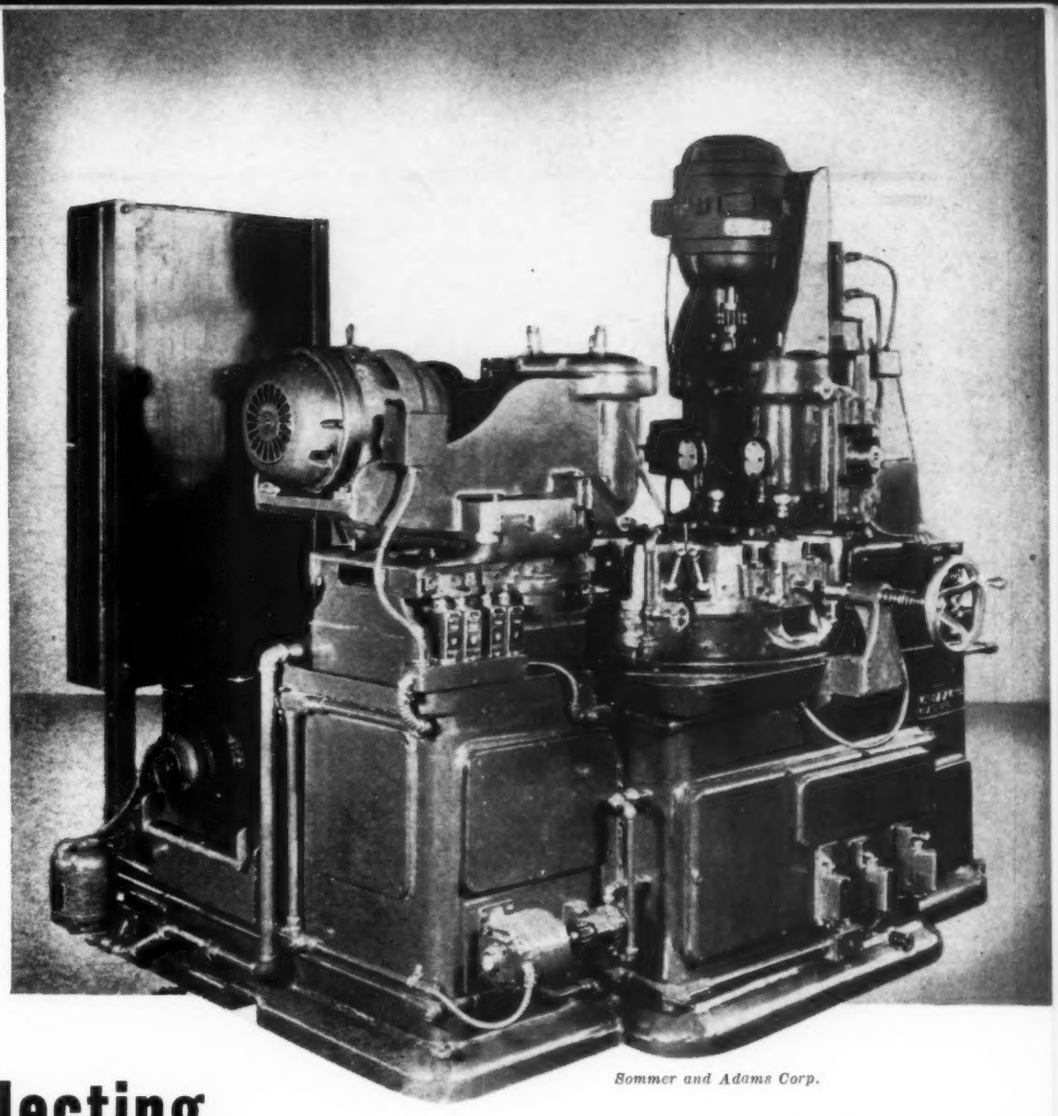


Fig. 1 — Limit switches and interconnected control elements provide automatic sequence operation for this 3-spindle milling machine.

Sommer and Adams Corp.

Selecting LIMIT SWITCHES

C. S. KIDD
Engineer, Allen-Bradley Company

Ten mechanical questions and five electrical qquestions that should be answered when applying limit switches to machines. A worked out example and six typical limit switch problems are solved with a diagram and explanation of each electrical circuit.

FOR MANY MACHINES the proper application of electrical limit switches is necessary for satisfactory performance and skillful designers always consider them to be integral elements when designing such machines. When complete layouts are made of machines with sequenced operations and the limit switches are merely after-thoughts, usually the designer must resort to "Rube Goldberg" linkages between limit switches and their actuating motion. For a neat, efficient, and accurate machine, it is always best to plan at the outset for the proper selection and application of the required limit switches.

A typical application of limit switches is shown in Fig. 1, where seven limit switches are integrated into the control of a three spindle milling machine. The actuating motions are transmitted to two central locations, the limit switches being grouped in front with four at the top left and three more at lower right. By planning for the limit switches at the time the machine was designed, the proper types were applied, the mounting space is adequate, and inspection and maintenance is simplified.

With a basic understanding of the types of switch mechanisms, motions, and operating linkages (PRODUCT ENGINEERING, Sept. 1946, page 132) the following questions are helpful in selecting the right limit switch for a desired function on the basis of mechanical and electrical requirements.

Mechanical Factors

1. What type of motion is involved? (Linear or rotary).

2. How much of this motion is available to operate the limit switch?

3. What is the speed and force of this motion?

4. What degree of accuracy is required in changing this motion?

5. How much overtravel or drift will this machine have after operating the limit switch?

6. How can this motion best be transmitted to the limit switch?

7. What type of lever will operate most efficiently with the transmitting means selected in question No. 6?

8. If rotary motion is required, how

many revolutions must be made before the contacts operate?

9. How much mounting space is available for the limit switch?

10. What type of inclosure should be used? (Such as general purpose, watertight, oil tight, or explosion proof.)

Electrical Factors

1. What type of contacts are required? (Normally open-normally closed, etc.)

2. Are maintained or spring return contacts required?

3. What load will the contacts be required to handle? (Inrush and sealed current values.)

4. How frequently will the contacts make or break the load in question No. 3?

5. Should slow or snap action contacts be used? (Use snap action when in doubt.)

To illustrate the use of these questions, the requirements of a machine will be described and limit switches will be selected to meet the required operating conditions.

A dry grinding machine is powered by three electric motors. One motor is coupled directly to the grinding wheel, another motor is coupled to a hydraulic pump, and the third motor is geared to the grinding head to raise and lower the grinding wheel in order to compensate for different sizes of work to be ground. Limit switches are required to stop the upward motion of the head before it reaches a fixed stop and to stop the downward motion before the grinding wheel reaches the work. The downward motion is motor driven until the limit switch is operated, after which the operator lowers the grinding wheel manually. The wheel is raised and lowered with a force of 30 lbs. at a speed of one-half ft. per second. The head will drift or coast 1/16 in. after the motor is disconnected. The work moves back and forth under the wheel on a hydraulically operated table.

Answers to the list of questions dealing with mechanical considerations are:

1. Only linear motion is necessary since the wheel is raised and lowered in a straight line.

2. Limited operating motion is available.

3. Thirty pounds force is available to operate the switch. This force is ample for any limit switch. A speed of one-half ft. per sec. is sufficient to operate the limit switch with a positive action.

4. Accuracy is important since damage to the machine bed, the work, and the grinding wheel, could be caused if the head did not stop its motion at the proper place.

5. Based on the described operating cycle, the machine drifts 1/16 in. after the motor is disconnected.

6. Small tapered projections can be mounted on the head to transmit the motion to the limit switches. The lower projection should be adjustable to allow for different sizes of material to be placed under the wheel.

7. A roller lever limit switch will lend itself to this application. Since accuracy is important a precision limit switch should be used. The switch should be constructed so that the rod can move at least 1/16 in. after the contacts operate. This will compensate for the drift of the head.

8. No rotary motion is used.

9. Mounting space is ample for two precision limit switches.

10. General purpose inclosures will be used since no sprays are involved.

Answers to the list of questions involving electrical consideration are:

1. Both limit switches are to stop the motion of the head therefore the motor circuit must be opened. Each limit switch must have one normally closed contact.

2. Spring return contacts should be used. The contacts should close as soon as the head moves away from the limit switch.

3. The limit switch contacts will carry the coil current of a magnetic reversing contactor. These coils require 0.71 amp. inrush current and 0.11 amp. sealed current.

4. The limit switches will operate only when the work to be ground is inserted or removed from the machine. Therefore, the limit switches will operate infrequently.

5. Snap action contacts should be used.

Summarizing, the required limit switches should be:

1. Roller lever operation,
2. Precision type,
3. General purpose inclosure,
4. One normally closed contact,
5. Spring return type,
6. Snap action contacts.

Based on the results of the analysis, Fig. 2 shows the layout and the wiring diagram for the limit switches applied to this dry grinding machine.

By following a similar question-and-answer analysis, the logic of the solutions of the different limit problems in Table I can be traced.

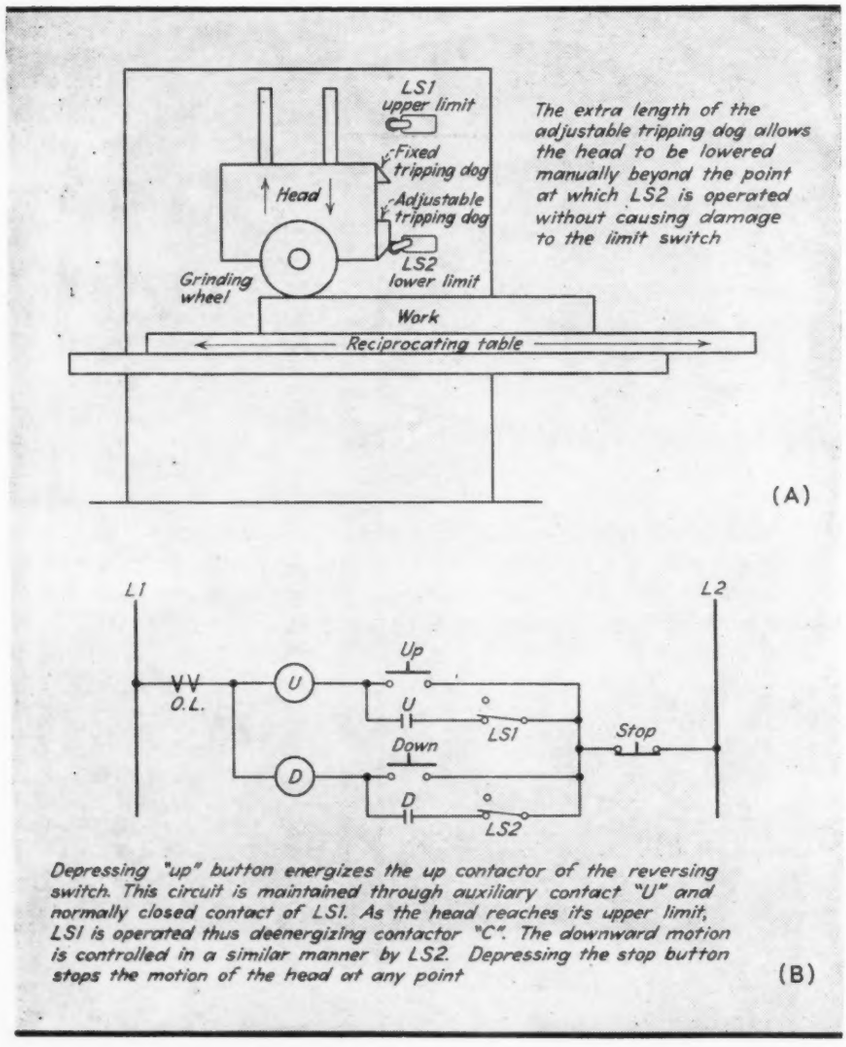

LS1
upper limit
Fixed
tripping dog
Head
Adjustable
tripping dog
LS2
lower limit
Grinding
wheel
Work
Reciprocating table
The extra length of the adjustable tripping dog allows the head to be lowered manually beyond the point at which LS2 is operated without causing damage to the limit switch
(A)
L1
L2
O.L.
U
Up
U
LS1
Stop
Down
D
D
LS2
Depressing "up" button energizes the up contactor of the reversing switch. This circuit is maintained through auxiliary contact "U" and normally closed contact of LS1. As the head reaches its upper limit, LS1 is operated thus deenergizing contactor "C". The downward motion is controlled in a similar manner by LS2. Depressing the stop button stops the motion of the head at any point
(B)

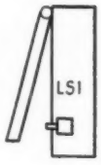
Fig. 2—A simple application of limit switches to a dry grinding machine. (A) Schematic layout of installation. (B) Diagram of the electrical control circuit.

Table I—Six Typical Limit Switch Problems and Their Solutions

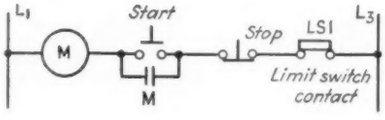
PROBLEM 1

CONDITIONS. Limit switch applied as a safety device so the machine cannot be started if the door to the operating mechanism is open.

REQUIRED LIMIT SWITCH. One push rod type, normally open contacts, slow action with the spring return.



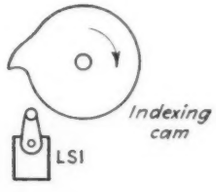
ELECTRICAL CIRCUIT. Start button energizes magnetic starter *M*, starting driving motor. The circuit is then maintained through auxiliary contact *M* until stop button is depressed or limit switch LS_1 opens when the machine door is opened.



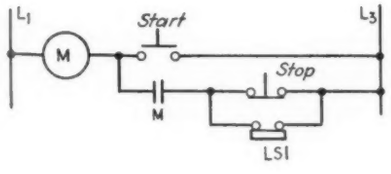
PROBLEM 2

CONDITIONS. To control a cam operated machine so it will stop only in a certain position. The machine should continue its movement until it reaches the desired stopping point, even though the stop button has been depressed.

REQUIRED LIMIT SWITCH. One roller lever type, normally closed contacts, snap action contacts, spring return.



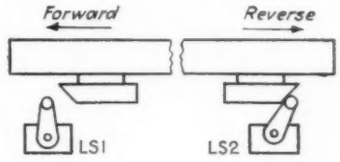
ELECTRICAL CIRCUIT. Magnetic starter *M* controls driving motor. To stop the machine, stop button is held depressed, the machine will continue to operate until the indexing cam has rotated to the stopping position, at which it opens the limit switch LS_1 and the machine stops.



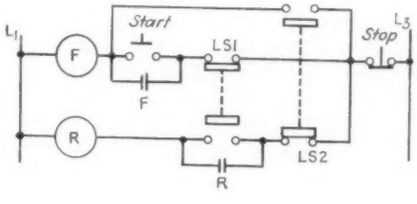
PROBLEM 3

CONDITIONS. To reverse the motion of a reciprocating table by reversing the motor driving the table. The reciprocating motion of the table is to continue until a stop button is depressed. The table is to always start in the forward direction.

REQUIRED LIMIT SWITCH. Two limit switches with roller levers, one normally open and one normally closed contacts each, snap action contacts, spring return.



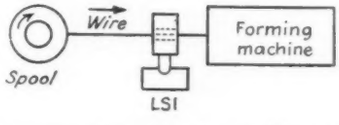
ELECTRICAL CIRCUIT. Start button energizes the forward magnetic contactor; circuit is maintained through interlock *F* and the normally closed contact of LS_1. As the table moves to a position where LS_1 is tripped, the forward contactor is deenergized; since the normally open contact of LS_1 has now closed, the reverse contactor *R* is energized and maintained through auxiliary contact *R* and the normally closed contact of LS_2. The table then reverses and moves in the reverse direction until limit switch LS_2 is tripped. The operation of LS_2 deenergizes the reverse contactor and again energizes the forward contactor. This will continue until stop button is depressed. Machine will always start in forward direction if both limit switches are in normal position.



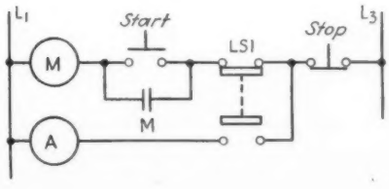
PROBLEM 4

CONDITIONS. In a wire-forming machine, if the wire becomes twisted or knotted, the machine should stop and sound a bell.

REQUIRED LIMIT SWITCH. One rod or chain type with an eyelet added to the lever. One normally open and one normally closed contact, snap action contacts, maintained contact type.



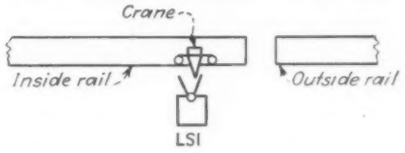
ELECTRICAL CIRCUIT. Starter *M* controls driving motor. Wire runs through eyelet of lever on limit switch and tangled wire will trip LS_1. When tripped, LS_1 stops the machine and bell, *A*, rings until trouble is corrected and LS_1 is reset by hand.



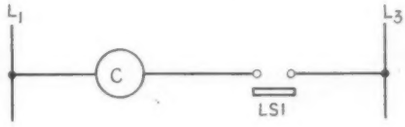
PROBLEM 5

CONDITIONS. An overhead crane operates both indoors and on an outdoor loading dock. The outside power rail is to be energized automatically when the crane reaches a certain position on the outward trip and deenergized when the crane reaches this point of the return trip.

REQUIRED LIMIT SWITCH. One fork lever type, one normally open contact, snap action contact, maintained contact type.



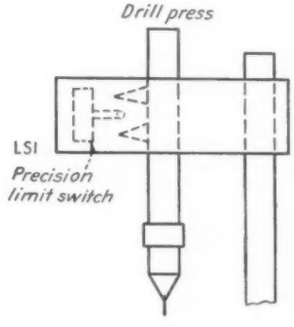
ELECTRICAL CIRCUIT. On outward trip the crane closes contacts at limit switch LS_1 and contactor *C* is energized; thus the outside power rail is energized and remains so until crane opens LS_1 on return trip.



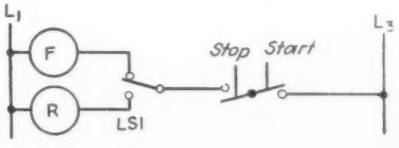
PROBLEM 6

CONDITIONS. Standard drill press used for tapping by reversing the rotation of the spindle at the ends of its travel.

REQUIRED LIMIT SWITCH. One roller lever type, one normally open and one normally closed maintained contact with snap action. Use precision type limit switch because of accuracy required and limited space.



ELECTRIC CIRCUIT. Motor is started in forward direction by depressing the maintained contact start button. At end of tapping feed LS_1 is tripped causing forward contactor to drop out and energizing reverse contactor, thus reversing the rotation. When the tap reaches the upper limit of its travel, LS_1 is restored to its original position and the tap again reverses. To stop machine, stop button is depressed.



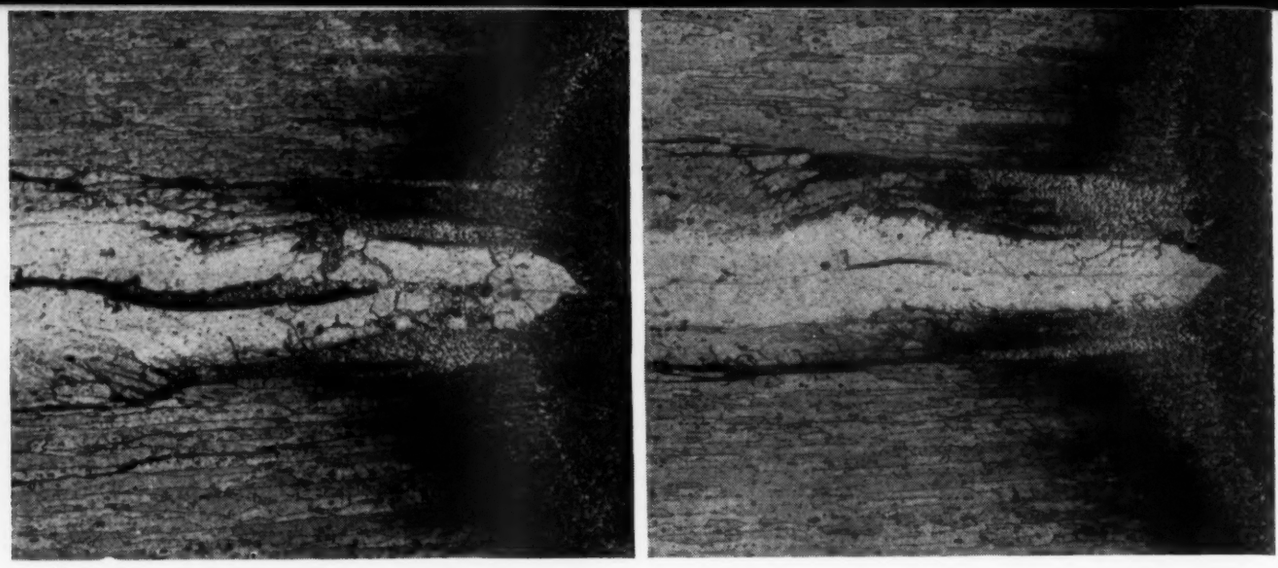


Fig. 1—Corona bonding, which accounted for about 16 percent of the shear strength of spot welded 75ST alloy, was damaged by salt spray as shown at the left. Compare the light, irregular fusion in corona area at right.

Effects of Corrosion On Spot Welded 75ST Alclad Alloy

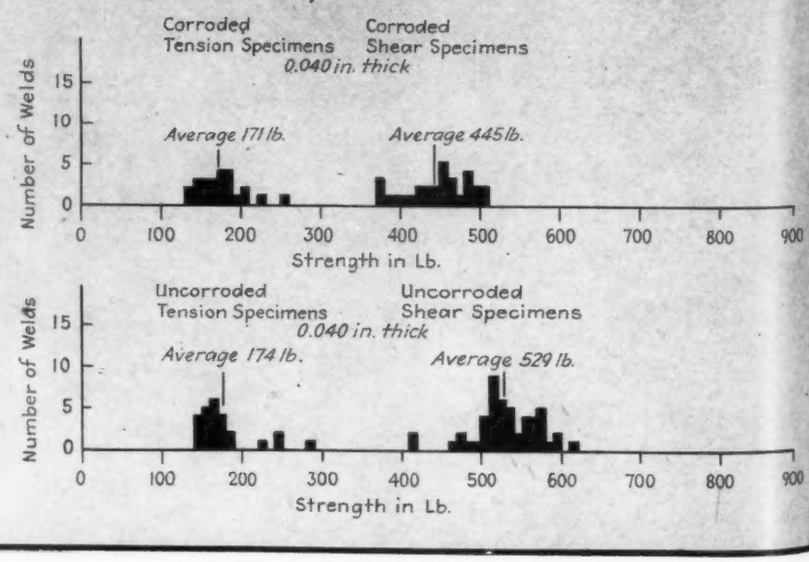
MARIO L. OCHIEANO
Production Methods Engineer, Lockheed Aircraft Corporation

Improper material preparation and welding techniques result in erratic bonding in the corona area of 75ST Alclad spot welds. This nonuniform bonding is affected by salt spray resulting in an inconsistent reduction of shear strengths. Results of tests showing these results, as well as data from tension specimens and data on the preparation and welding of specimens, are discussed.

SALT SPRAY attacks light and irregular corona bonding of spot welds in 75ST Alclad alloy to reduce the shear strengths of the welds. Examination of corroded spot welds revealed that the degree of the corrosive attack or penetration into the corona area was not uniform even on specimens that were prepared and spot welded under identical conditions. Salt spray causes corrosion of the corona area of some specimens only slightly, or in spots, while on others it attacks the entire corona bond. These are some results of tests conducted by the author to determine the spot weld characteristics, welding technique and corrosion aspects of 75ST alloy.

Corona bonding and the effects of corrosion upon it are shown in Fig. 1. A light, irregular fusion is present throughout the corona area of the uncorroded specimen. On the corroded specimen the fused clad coating within the corona area is broken and damaged. Corona bonding in the 75ST alloy accounted for about 16 percent of the developed shear strength of the spot weld. When the corona bonding suffers damage

Fig. 2—Shear and tensile strengths of spot welds in 75ST specimens of 0.040 in. in thickness before and after salt spray corrosion.



through corrosion, the shear strengths are dimished accordingly.

Average strength losses of shear specimens subjected to 250 hours of salt spray were as follows:

Material thickness, in.	Average strength loss, percent
0.025 and 0.025	3.3
0.065 and 0.064	16.1
0.040 and 0.040	15.9

Examination of the sheared specimens showed no evidence that the salt spray penetrated to or attacked the weld nugget. This not only is in agreement with the remarks of the foregoing paragraph but also is supported by the results of tension tests wherein the specimens subjected to salt spray showed no significant loss in tensile strength over those tested in the uncorroded condition. Table I contains data from the tests of corroded tension specimens. The right-hand column shows the tensile strength compared with equivalent specimens that were not subjected to salt spray.

Three thicknesses of 75ST aluminum sheet were used in making the spot weld specimens; namely 0.025, 0.040 and 0.064. The corresponding widths of the shear test specimens were 5/8, 3/4 and 1 in., respectively. Two pieces to be joined were overlapped by 3/4 and 1 in. and the spot weld centrally positioned with respect to the overlapped area. The average weld button diameters were 0.14, 0.16 and 0.22 in. for each of the three thicknesses of materials.

Typical shear test data are given in Table II, the right-hand column showing

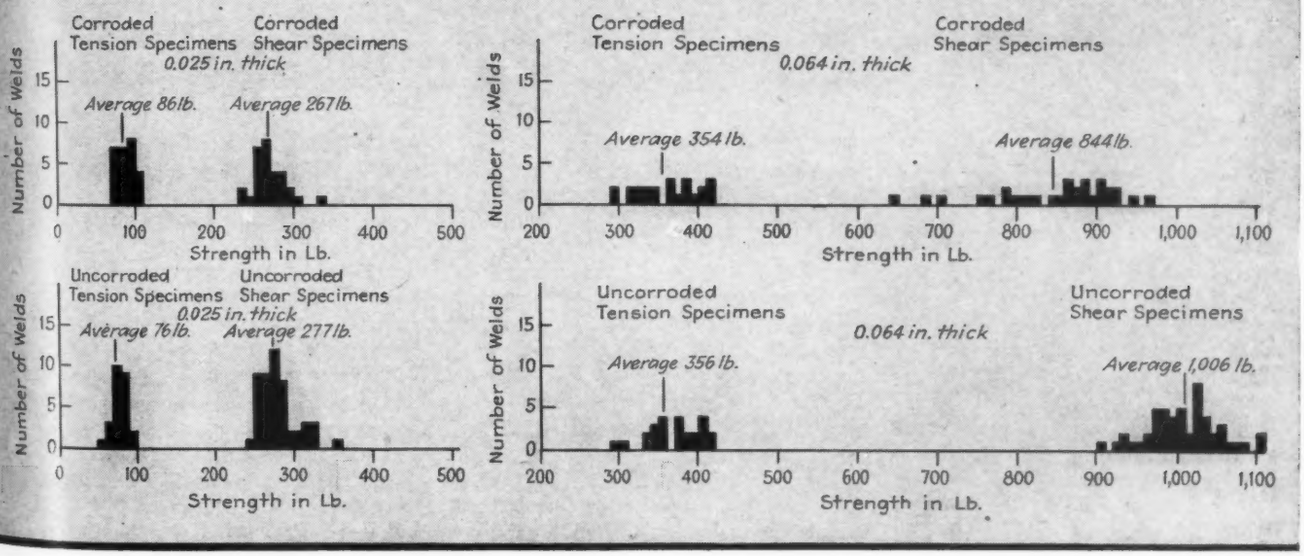
Table I—Typical Test Results of Tensile Specimens After 250 Hours in Salt Spray

Material thicknesses, in.	Ultimate strength, lb. max.	min.	avg.	Deviation from avg., percent	Number specimens tested	Ratio of tension to shear strength, percent	Ratio tensile strength of corroded to uncorroded, percent
0.025 and 0.025	105	70	86	+18.1 −18.6	26	32.2	+11.5
0.040 and 0.040	255	130	171	+33 −24	24	38.4	— 1.8
0.064 and 0.064	415	260	354	+14.7 −26.5	24	41.9	— 0.6

Table II—Shear Test Data of Specimens After 250 Hours in Salt Spray Compared with Uncorroded Specimens

Material thicknesses, in.	Ultimate strength, lb. max.	min.	avg.	Deviation from avg., percent	Number specimens tested	Number specimens within 10 percent of avg.	Ratio shear strength of corroded to uncorroded, percent
Corroded shear specimens							
0.025 and 0.025	335	235	267	+20.3 −12.6	30	26	— 3.3
0.040 and 0.040	505	370	445	+11.9 −16.9	30	23	—15.9
0.064 and 0.064	965	640	844	+12.5 −24.1	29	23	—16.1
Similar but uncorroded shear specimens							
0.025 and 0.025	350	250	277	+20.9 — 7.6	50	42	—
0.040 and 0.040	595	415	529	+11.1 −20.0	50	44	—
0.064 and 0.064	1100	900	1006	+ 8.5 −10.5	50	49	—

Fig. 3—Shear and tensile strengths of spot welds in 75ST specimens of 0.025 in. in thickness before and after salt spray corrosion. Fig. 4—75ST specimens of 0.064 in. in thickness.



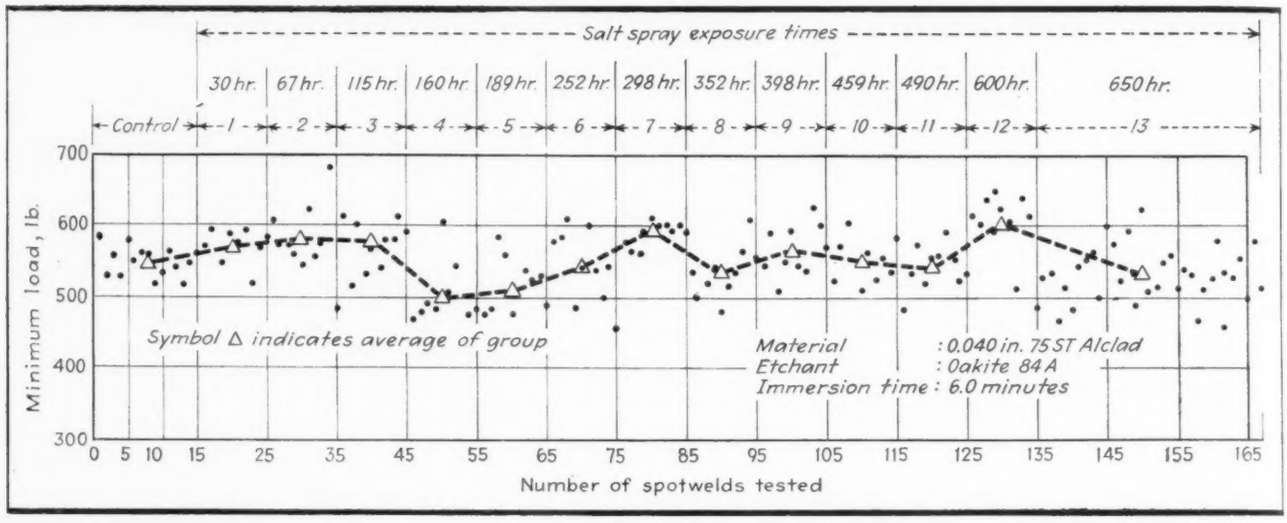


Fig. 5—Data from shear tests showing dispersion of shear strengths in 75ST specimens of 0.40 in. in thickness.

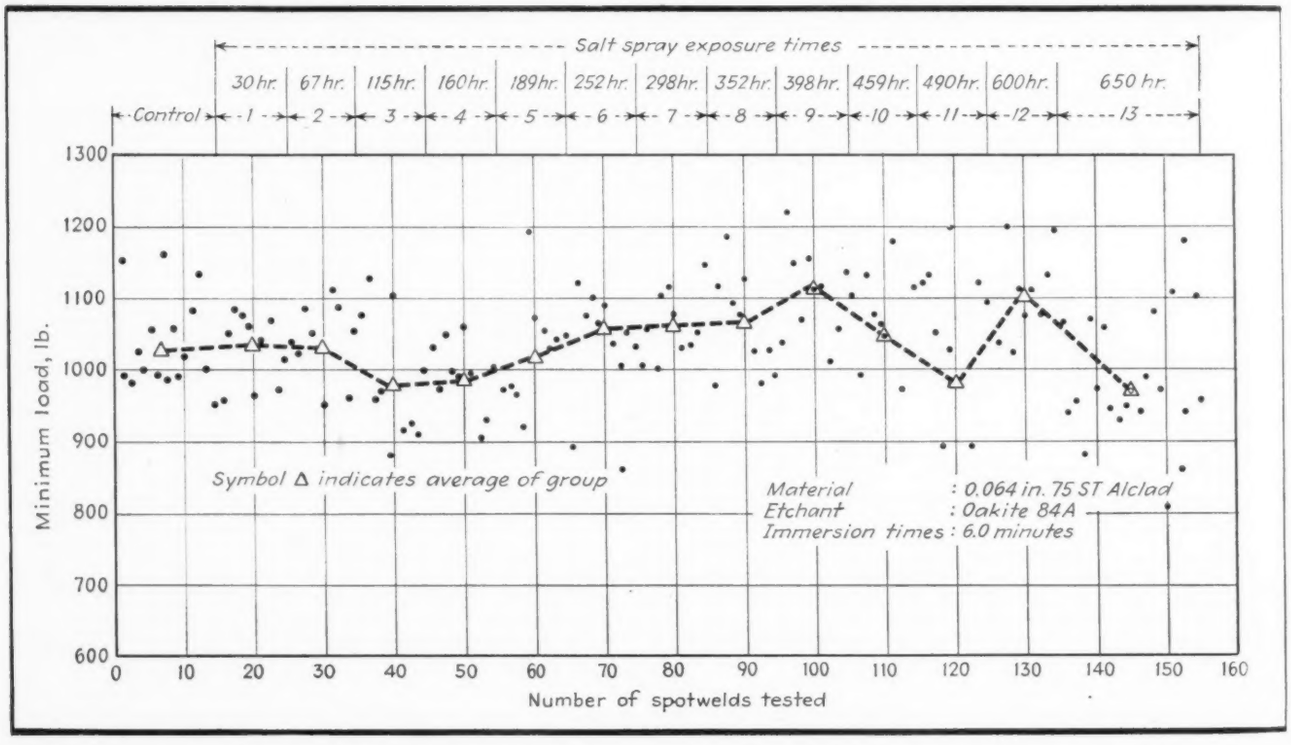


Fig. 6—Data from shear tests showing dispersion of shear strengths in 75ST specimens of 0.064 in. in thickness.

the comparison between the shear strengths of corroded and uncorroded specimens. Figs. 2, 3 and 4 are dispersion charts comparing test results of both shear and tensile specimens that had been and had not been subjected to salt spray.

Of significance is the fact that the tensile strengths of the 0.040 and 0.064 gages show only slight variations between corroded and uncorroded specimens, whereas the shear specimens exhibit a market reduction in strength and consistency. Average test results are indicated, although minimum strengths may be considered by some as a better index of allowable strengths.

Besides the specimens subjected to 250 hours of salt spray, groups of spot weld specimens were tested after 50-hour intervals of salt spray up to 650 hours. Figs. 5 and 6 show the results of two such tests, one being for 75ST sheet of 0.040 in. and the other of 0.064 in. in thickness. These specimens were prepared for welding by cleaning in a soap solution of 6 oz. per gal. for 10 min. at 180 deg. F. Then they were immersed in Oakite etching fluid, 84A, at 180 deg. F. for 6 min. The specimens were spot welded within four hours after cleaning.

The data on the strength dispersion charts, Figs. 5 and 6, indicate that long material preparation times and short atmospheric exposure times before welding are essential for successful welding of the 75ST Alclad alloy. Another condition, which is favorable to producing a good corona bond, is a low, uniform sheet contact resistance. The critical nature of preparation of 75ST specimens for spot welding justifies the use of apparatus to control the sheet contact resistance within set limits. Use of such equipment permits the immediate detection of contaminated cleaning solutions and improper immersion times, and consequently it can be used as an indicator of up to standard spot welds. Other data from the tests were published in the September number of PRODUCT ENGINEERING.

Sensitive Ball Bearing Test Unit Measures Small Changes in Torque

J. L. SNEAD and H. GISSER
Frankford, Arsenal

Testing device uses electrical resistance strain gages to determine the frictional characteristics of high precision ball bearings. Test unit can be used to obtain data for selecting new ball bearings, for studying bearing lubricants, and for measuring changes in operating torque of ball bearings from corrosion during storage.

FRICTIONAL CHARACTERISTICS of high precision ball bearings can be automatically recorded in a short period of time with a recently developed ball bearing test unit. The test method is nondestructive and measures changes in torque as small as 0.06 gram-centimeter.

Although the unit was developed primarily for determining the changes in the operating characteristics of precision ball bearings that occur during exposure or use, it is applicable for selecting unserviceable ball bearings from a manufactured lot. Other applications for the device are the study of lubricants for ball bearings, and the measurement of changes in the operating characteristics of ball bearings caused by corrosion while in storage.

The test unit was designed especially to study the performance of bearings that must operate smoothly in service with a maximum applied torque of approximately 5 gram-centimeters. To allow for the frictional effects caused by the traces of corrosion that develop in assembly, storage and service, the bearings when new are required to operate satisfactorily at applied torques not greater than 1 gram-centimeter.

The torque measuring elements in the tester are four type A-14 (500 ohm) SR-4 electrical resistance strain gages. Two gages are mounted on each side of a strip of spring bronze 4-1/2 × 5/8 × 0.013 in. as shown in Fig. 1. The bottom end of the strain spring, to which an angle bracket is fastened, is bolted rigidly to the floor of the test unit. The spring thus forms a vertical cantilever beam.

During testing the bearing is held in a holder made of brass strip, which presses tightly over the outer bearing race. The holder has two projections as shown in Fig. 2. With the bearing and holder pressed on the end of the drive motor shaft and the motor in operation, the torque of the frictional forces in the bearing is transmitted to the free end of the spring cantilever beam by a projection on the holder. The beam is thus deflected and the linear deformation loads the resistance strain gages. Fig. 3 shows the assembly of the bearing test unit.

The change in resistance of the strain gages as they are deformed by the deflection of the spring beam is recorded in terms of current flow by a milliammeter in an electrical circuit, Fig. 4, that includes the four gages connected as the arms of a resistance bridge, an oscillator, a bridge balancing unit, and an amplifier.

CALIBRATION. In calibrating the test unit, the bridge is first electrically balanced without ball bearing and holder. The unit is then placed on its side so that the spring is horizontal. Then a wire hook is suspended from the spring, and the weight of the spring and hook is counterbalanced,

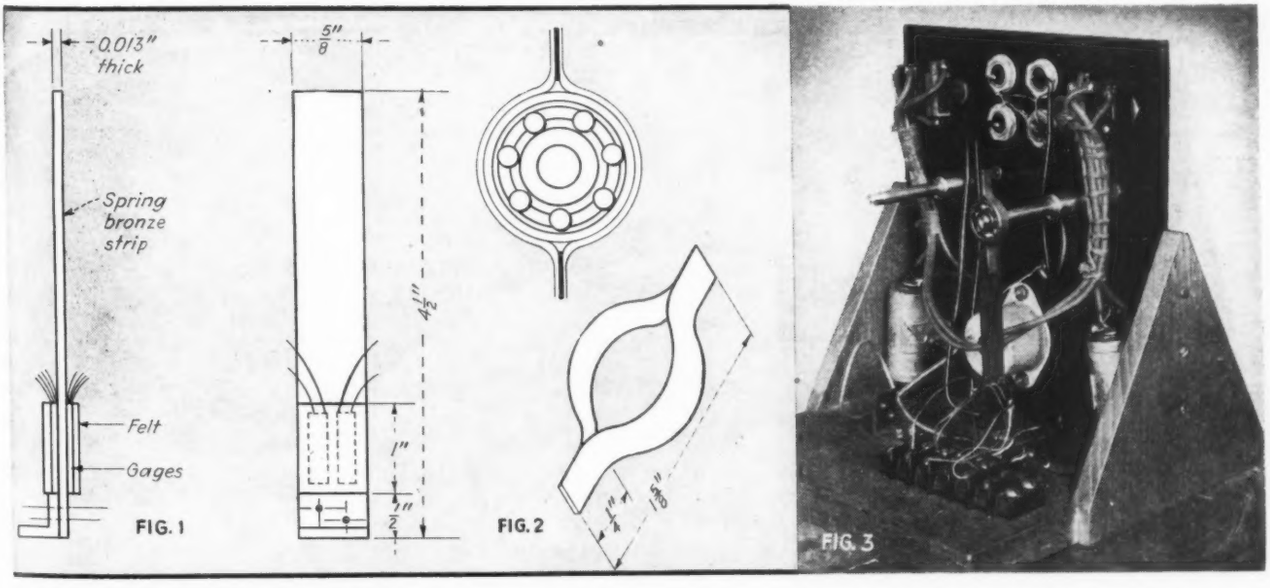


Fig. 1—Method of attaching electrical resistance strain gages to spring strip. Fig. 2—Detail of bearing holder and ball bearing encased in holder. Fig. 3—Assembly of bearing test unit with bearing ready for test.

Table I—Calibration Data

Weight, milligrams	Current milliamperes	Standard deviation, 10 readings
100	0.12	0.0454
200	0.28	0.0595
300	0.40	0.0768
400	0.59	0.0467
500	0.74	0.0458
600	0.90	0.0329
700	1.07	0.0471
800	1.24	0.147
1,000	1.58	0.0451

as shown in Fig. 5, with sufficient weight to rebalance the bridge electrically. Weights are then hung from the spring and the resulting milliammeter readings recorded as shown in Table I. These data are then fitted to a straight line by the method of least squares. For the data given in Table I, where

I = current, milliamperes
W = weight, milligrams

The equation for the calibration curve is

$$I = 0.0016\,W - 0.0525$$

By reference to Fig. 6 it will be seen that the calibration curve departs from linearity when the weight on the particular spring being calibrated is more than 800 milligrams, which corresponds to currents greater than 1.25 milliamperes.

Operation. In testing, the unit is set upright and the bridge balanced. A bearing with holder is put on the motor shaft, the motor started, the unbalanced current amplified, and the amplifier output recorded on a recording d.c. milliammeter. The gain of the amplifier is checked periodically by shunting one of the gages with a calibration resistor built into the bridge balancing unit and determining the current output caused by unbalance, and adjusted when required.

Typical data obtained from two standard No. 37 precision ball bearings are shown in graphs *A* and *B* of Fig. 7. The data recorded in the graphs are for counterclockwise and clockwise revolutions, both before and after being subjected to salt spray for one hour. Each curve represents one revolution, at 1 r.p.m. of the bearing. The zero lines on the graphs were drawn in after the data were recorded.

The difference in torque required to turn the bearings before and after corrosion, in most instances, is so small in magnitude that it cannot be readily detected by hand rotation. The graphs clearly indicate the differences among new bearings. Bearings *A* and *B* appeared to be equivalent when turned by hand, but when tested showed widely different frictional characteristics. Bearing *B* has much poorer characteristics.

The test data on bearings *A* and *B* show that the frictional characteristics may be different when the bearing is rotated in different directions. The difference in characteristics is not caused by entrance or exit of dirt after the first two revolutions, because the clockwise and counterclockwise revolutions were alternated during all tests.

The differences in the frictional characteristics of two No. 37 bearings (*C* and *D*, Fig. 7) specially processed for use in synchronous units and standard No. 37 bearings are apparent from the graphs. The average torque of bearings *C* and *D* is 0.17 gram-centimeters. The sensitivity of the bearing tester is approximately 0.05 milliamperes, which corresponds to a torque of 0.056 gram-centimeters.

General Discussion. In assembling the unit, difficulty was at first experienced with the mechanical system. Since the torques encountered in synchronous unit bearings are unusually small, the spring must be quite thin to obtain appreciable strain. On the other hand, decreasing the thickness of the spring decreases the sensitivity of the gage. At a spring thickness of 0.008 in. it was found that the lead wires were sufficiently stiff to cause maximum spring strain in the area above the strain gages. The first lead wires used were No. 20 stranded hookup wire (synthetic insulation) but these were later changed to a double silk covered especially flexible wire.

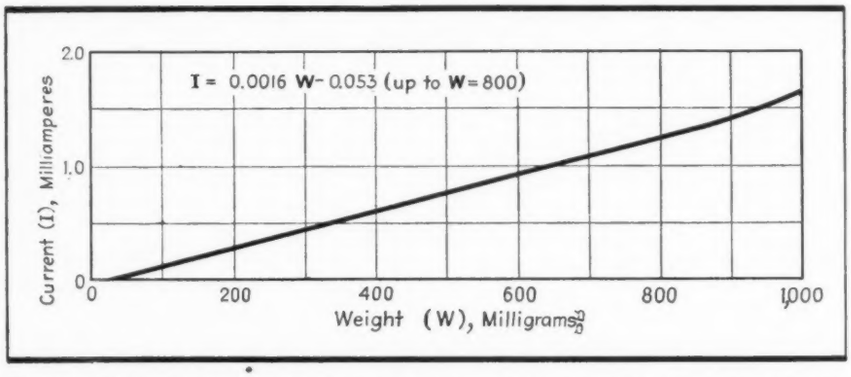


Fig. 6—Calibration curve that correlates current flow and load on spring beam.

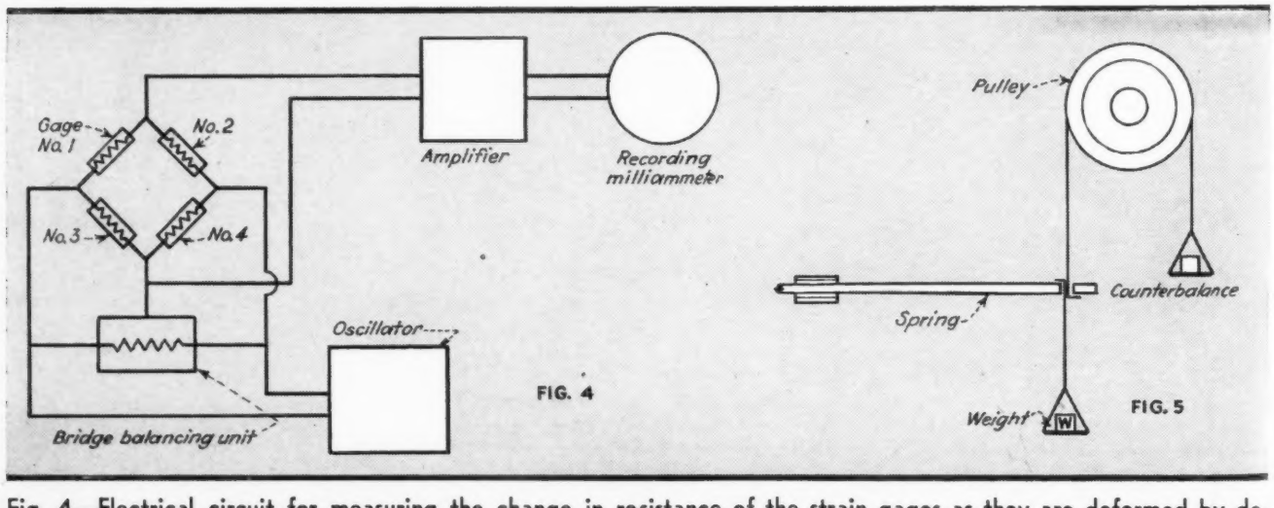


Fig. 4—Electrical circuit for measuring the change in resistance of the strain gages as they are deformed by deflections of the spring beam. Fig. 5—Method of calibrating the bearing test unit.

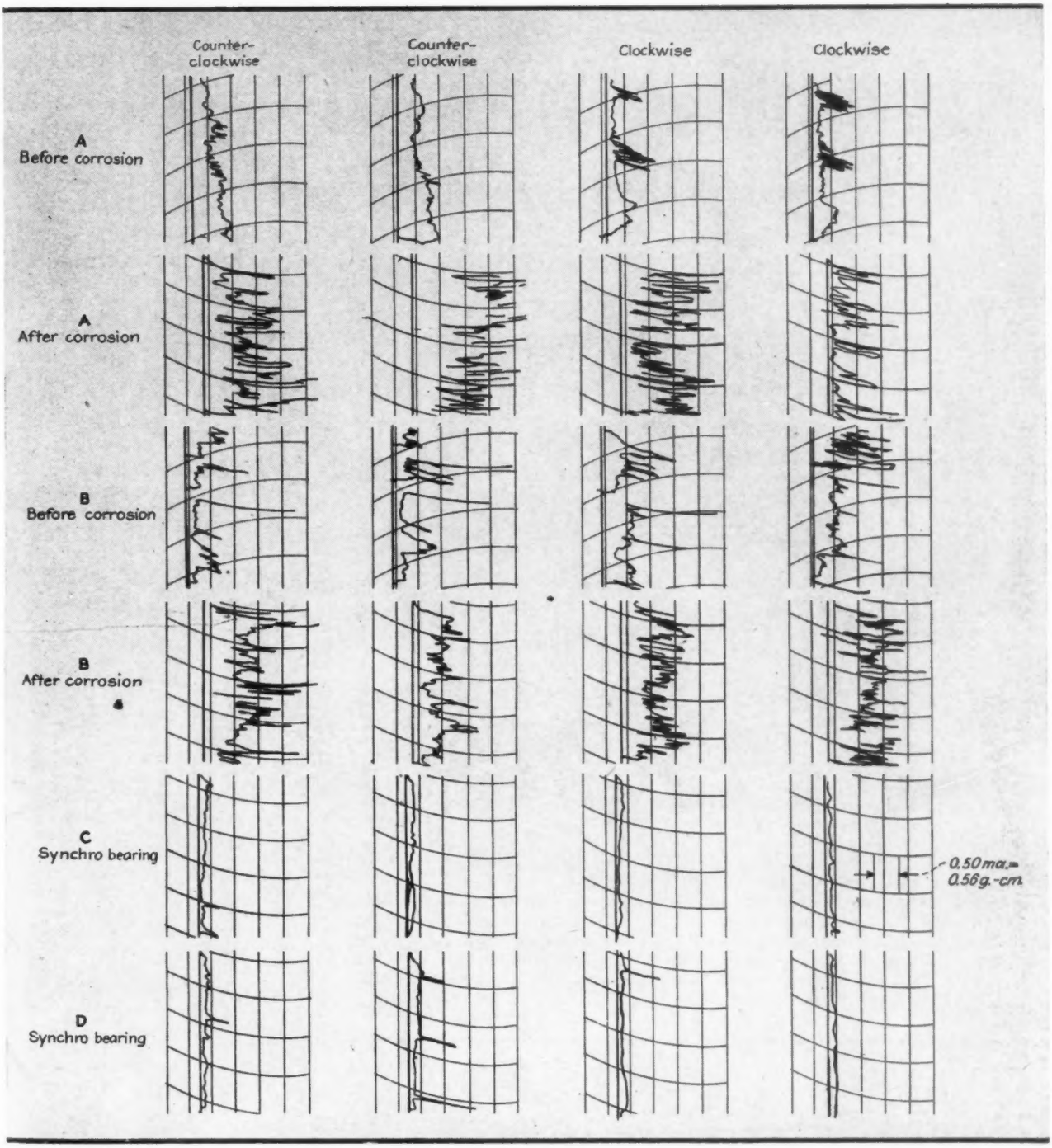


Fig. 7—Milliammeter readings for four bearings during torque tests. (A) and (B) Two Standard No. 37 bearings before and after exposure to salt spray. (C) and (D) Two No. 37 bearings specially processed for synchronous units.

But even with this change it was necessary to increase the spring thickness, since the lead wires still contributed to stiffness in the gage area.

At first only two strain gages were used, one on each side of the spring. But later four gages were used to increase the sensitivity. The material used to protect the gages from injury was also found to exert an appreciable effect. A thin flexible Vinylite sheet used at first was abandoned in favor of a flexible felt strip.

A more rigid system resulting from the use of a thicker spring was also considered to avoid damage in handling.

A new portable bearing test unit has been designed having replaceable springs and spring holders for different size bearings. The new unit is inclosed in a box to prevent errors caused by slight drafts or temporary changes in atmospheric temperatures in the neighborhood of the equipment, and to make the unit more resistant to injury by shock or mishandling. In the new unit, spring assemblies and bearing mounts are interchangeable.

The authors have found strain gages useful in measuring the smoothness of operation of delicate instrument movements and are now studying a recording dynamometer, using strain gages as the sensitive elements, for measuring increased torque and grittiness that develops in delicate instruments during storage.

ACKNOWLEDGEMENT: The authors are indebted to Dr. T. A. Read and Mr. J. L. Everett for suggestions during the development of the bearing test unit.

INDUSTRY AND SOCIETIES

RESEARCH—STANDARDS—PATENTS

Proposed Air Engineering Development Center

Fluid Dynamics Study To Be Specialty At New Site

ARMY AIR FORCES SCHEMES for a multi-billion dollar research program in guided missiles, jet engines, rockets, atomic propulsion and supersonic aircraft are embodied in plans for a new "Air Engineering Development Center." First announced several months ago (see July "Research Notes") by General Curtis E. LeMay, Deputy Chief of Air Staff, the program is still tentative, and no action can be taken until Congress votes on it at its next session, which begins in January.

As described by Gen. Le May, the proposed center is to include research on Fluid Dynamics, Thermodynamics, Structures and Materials, Electronics and Wave Phenomena, Physiology, Fuels and Propellants, and Instruments. The proposed center, which is being submitted for review to the aircraft industry and the National Advisory Committee for Aeronautics, is looked on by the Air Forces as a link between the pure science research of NACA and the practical application of industry. The A.A.F. feels that private organizations cannot afford the expenditure of the huge sums required for development and testing of modern aircraft. Present facilities at Wright Field are viewed as wholly inadequate.

Physical facilities of the proposed center, which will probably be located in the Far West, will represent an initial expenditure of $300,000,000. Initial estimate of the connected load for the center is approximately 1 million hp., with a maximum demand of approximately 500,000 hp. Because of the magnitude of this requirement and the various load conditions under which the facilities will operate, it is practically impossible to expect private power interests to handle such an enterprise. It would seem that such requirements can only be met by the use of existing or new generator equipment, at a government-owned and operated hydroelectric power site.

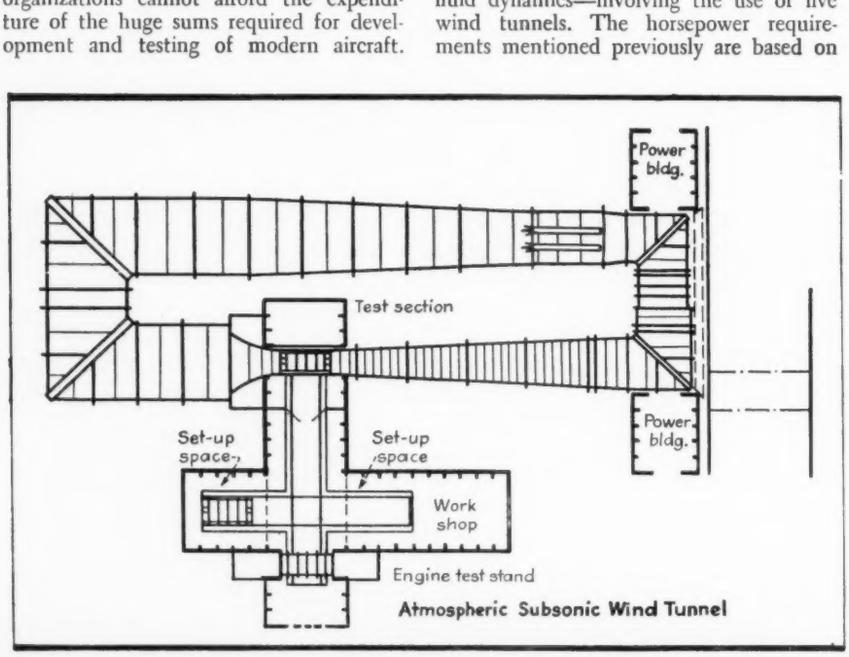
Fluid Dynamics

Most detailed section of the Air Forces proposal for a research and development center is that dealing with the study of fluid dynamics—involving the use of five wind tunnels. The horsepower requirements mentioned previously are based on simultaneous operation of two tunnels.

A survey of present fluid dynamic test facilities shows them to be entirely inadequate for the future development of long range, high speed missiles and aircraft. There are few tunnels in the United States at the present time which can be considered as being of possible use for development of this nature. The most powerful wind tunnels are the Wright Field Twenty Foot Tunnel and Ten Foot Tunnel, each utilizing 40,000 horsepower. The Twenty Foot Tunnel is a sea level tunnel having a speed of 450 miles per hour; while the Ten Foot Tunnel can be operated under altitude conditions and at speeds up to nine tenths the velocity of sound. The largest high speed subsonic wind tunnels are the NACA Sixteen Foot Tunnel having a speed of approximately 550 miles per hour and the NACA Eight Foot Tunnel which has a speed of approximately 650 m.p.h. The only wind tunnel capable of testing propulsion units under altitude and temperature conditions is the NACA Twenty Foot Tunnel having a speed of around 500 m.p.h. and attaining this speed only at the low density and temperature corresponding to 30,000 feet altitude. A few supersonic tunnels exist but these are small, having test sections under twenty four inches in diameter, and relatively slow, having maximum Mach Numbers (Mach Number 1 = speed of sound) of three and one-half. There are no facilities available at present for the testing of turbojets, rockets or their component parts under altitude and high Mach Number conditions. As an initial step in achieving leadership in the field of fluid dynamics, facilities for testing four specific speed ranges are recommended. These ranges are the Subsonic, Transonic, Supersonic and Hypersonic. The normal boundaries of these ranges are as follows:

1. Subsonic—Mach Number 0 to Mach Number 0.7
2. Transonic—Mach Number 0.7 to Mach Number 1.8
3. Supersonic—Mach Number 1.8 to Mach Number 5
4. Hypersonic—Mach Number 5 to Mach Number 10 or more.

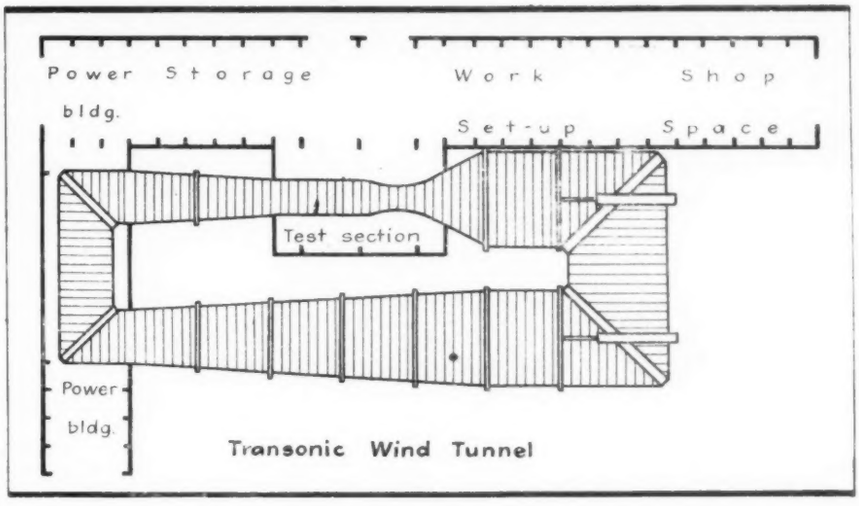
Although a great deal of work has been



Atmospheric Subsonic Wind Tunnel

done in the subsonic range, many types of military aircraft will continue to operate at these speeds and all types will have to pass through this region in attaining the higher speed ranges. The latest fighters have already edged into the transonic region where marked changes in the aerodynamic characteristics of the planes are known to occur. Very little is known of the fluid dynamics of both the supersonic and hypersonic ranges although missiles have been built and flown at supersonic speeds.

The following wind tunnels are proposed in order that these conditions of testing can be satisfactorily studied.



Transonic Wind Tunnel

ATMOSPHERE SUBSONIC WIND TUNNEL. In the development of new equipment a great deal of unrefined, shakedown type of testing is necessary. This requires moderately high speed tests where lift, drag, moments, and pressure distribution data can be obtained. Such tests do not require the use of an elaborate high speed, altitude type of wind tunnel with its array of temperature, pressure, and humidity equipment.

To serve this function at the Development Center a simple air exchanger type wind tunnel is proposed. The test section diameter should be between twenty and thirty feet and speeds up to at least nine-tenths the speed of sound should be obtainable. This tunnel would require approximately 100,000 hp. for its operation.

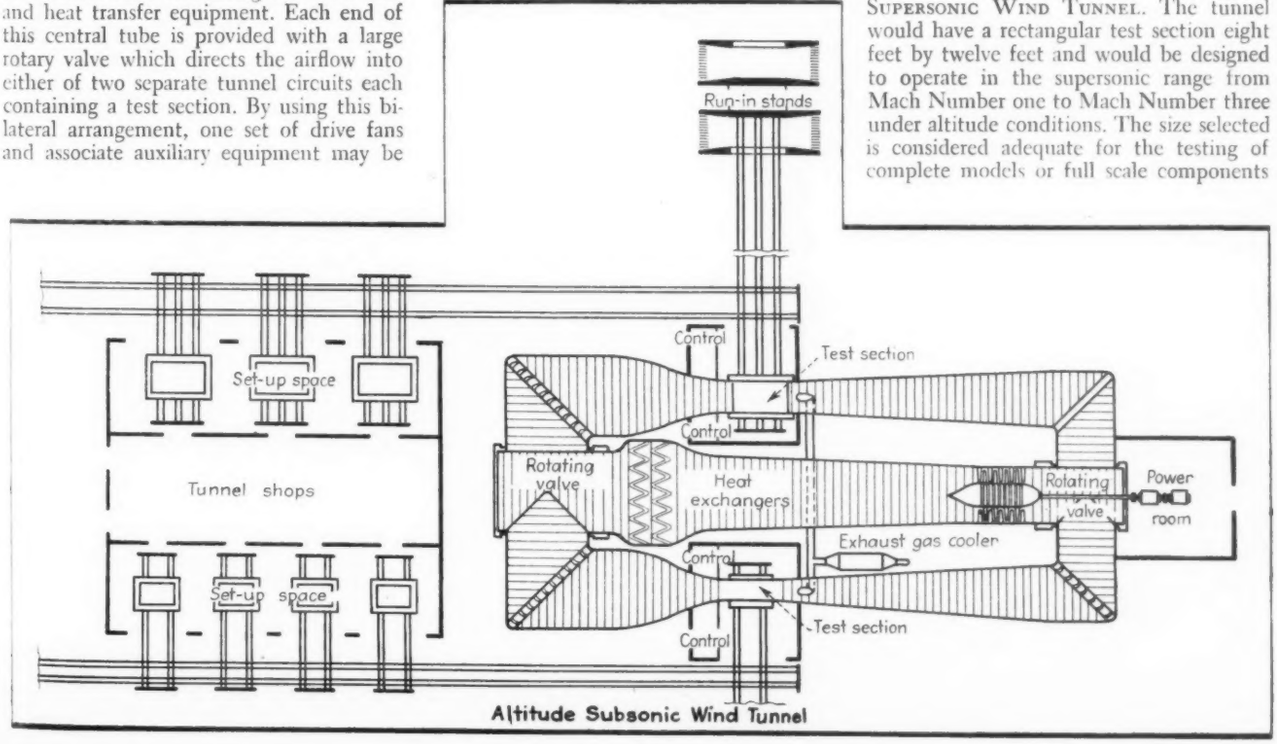
ALTITUDE SUBSONIC WIND TUNNEL. This wind tunnel would be used for testing full scale nacelles, thermal jet engines, engine propeller combinations, and for special propeller tests, up to a Mach Number of nine tenths with temperatures and pressures from sea level to 80,000 feet altitude. It is a bilateral tunnel, consisting of a center tube containing the drive fans and heat transfer equipment. Each end of this central tube is provided with a large rotary valve which directs the airflow into either of two separate tunnel circuits each containing a test section. By using this bilateral arrangement, one set of drive fans and associate auxiliary equipment may be made to serve two different size test sections. This permits operation of one test section while a test set-up is being made in the other. It is contemplated that one test section would be twenty feet in diameter and the other thirty feet in diameter.

The capacity of the refrigeration, air drying, air compressor, exhausting, and scavenging equipment should be sufficient to permit continuous operation of either test section.

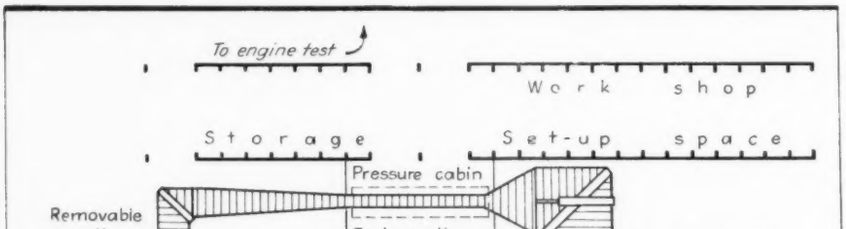
The test sections in each circuit of the tunnel should be mounted on tracks which would permit making a complete set-up of a test section in the shops and checking it on "run-in" test stands before placing it in the tunnel. The maximum utilization of the tunnel and its facilities would be obtained by providing two or three test sections of each size, each section being adapted to a particular type for test. These operating conditions would require about 120,000 shaft hp. to the fans.

TRANSONIC WIND TUNNEL. It is proposed that this tunnel cover the transonic range from Mach Number seven tenths to Mach Number one and eight tenths. This tunnel will be utilized for the study and development of full scale gas turbines, jet engines, propellers, and complete winged missiles for operation in the transonic speed range. It will be an altitude tunnel having a twenty foot diameter test section and will utilize the same scavenging, evacuation and refrigeration machinery as the altitude subsonic wind tunnel. It will be necessary to drive this tunnel with an independent compressor system powered with approximately 150,000 horsepower.

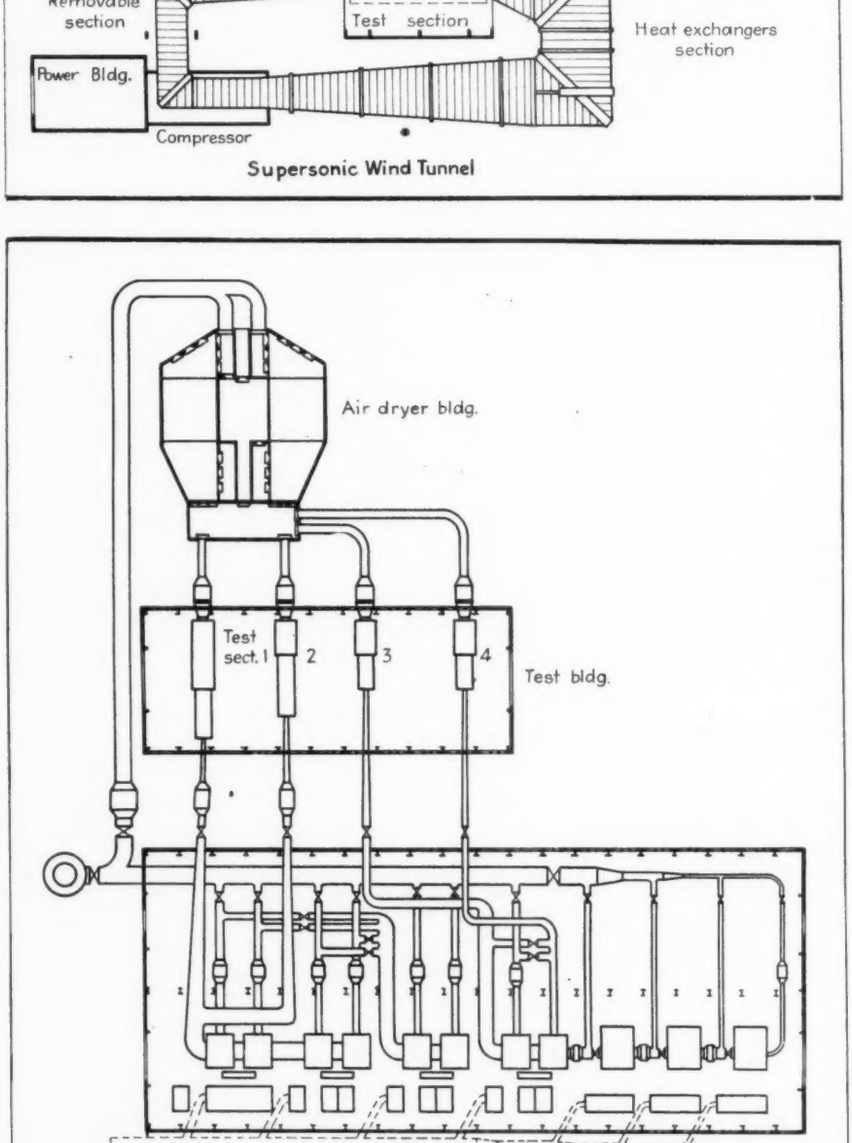
SUPERSONIC WIND TUNNEL. The tunnel would have a rectangular test section eight feet by twelve feet and would be designed to operate in the supersonic range from Mach Number one to Mach Number three under altitude conditions. The size selected is considered adequate for the testing of complete models or full scale components



Altitude Subsonic Wind Tunnel



Supersonic Wind Tunnel



Hypersonic Wind Tunnel

of ram jet, turbo-jet or rocket propelled missiles and aircraft. It will be capable of obtaining both aerodynamic and thermodynamic test data on these devices.

The design would be such as to permit operation at pressures greater than atmospheric so as to extend the range of Reynolds Numbers which can be attained. Test section pressures from one-sixth to two and one-half atmospheres are planned. For testing propulsion units the return circuit of the tunnel is removable permitting the exhaust gases to be discharged directly into the atmosphere.

The tunnel air stream would be driven by a multistage axial flow compressor with adjustable guide vanes for performance control. Characteristics of the machine should be such that operation would be at peak efficiency for most of the tunnel testing range. The compressor would require a prime mover of 150,000 horsepower.

This range and size of tunnel has been chosen as an engineering compromise between size, and power requirements and compressor performance.

HYPERSONIC WIND TUNNEL. It is proposed that this tunnel have a forty inch square test section and be capable of testing continuously at speeds corresponding to Mach Numbers from one to ten, thus fully covering the supersonic and entering the hypersonic range. Its primary purpose would be for the development of long range missiles and aircraft and their propulsion systems. Because of the small size test section, tests will, in most cases, be confined to scale models. Both aerodynamic and thermodynamic measurements will be possible.

The tunnel will be driven by a system of compressor stages, arranged to operate in various combinations, making it possible to obtain the complete range from Mach Numbers one to ten.

The power plant and compressor system would consist of several stages of compressors which could achieve an overall compression ratio 875 to 1 when operating against a pressure of 1.4 millimeters of mercury absolute. It is estimated that about 75,000 horsepower will be required to operate this compressor system.

A large tunnel having these characteristics would be desirable but the amount of power required and the complexity of equipment led to this selection.

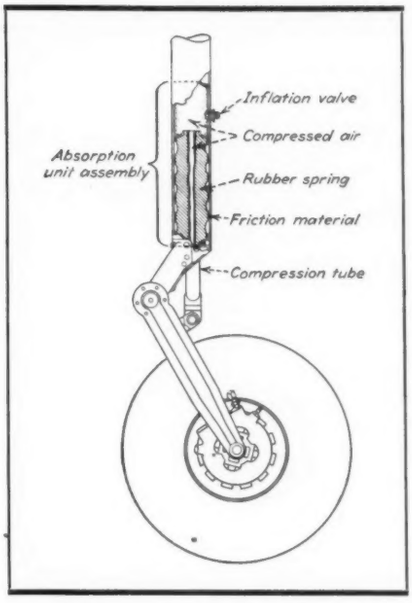
AAF Official Photo

This single-place flying-wing fighter plane, known as the XP-79B, is jet-propelled and is capable of attaining speeds better than 500 miles per hour. One distinctive feature of the plane, built by Northrop, is that the pilot flies the aircraft while lying in a prone position. It is an all-metal pressurized flying wing powered by two Westinghouse model 19B jet units. Twin fins are mounted on top of the wing and rudder assist ducts aid in operating split flap rudders at the wing tips. Its best operating altitude is around 25,000 ft.

WASHINGTON NOTES

Office of Technical Services In Commerce Department Set Up By Wallace

Henry Wallace, even after facing a policy defeat and removal from office, seems to be getting his way on at least one matter. Despite Congressional failure to take action on S. 1248, a bill sponsored by Senator J. W. Fulbright of Arkansas, which would set up an "Office of Technical Services" in the Commerce Department, an organization operating under that identical title has been established, and is now functioning in the Commerce Dept., with a budget allowance of $3,600,000 for the fiscal year beginning July 1, 1946. Embodying essentially the same program outlined in the Fulbright bill (see the May "Washington Notes" column for a complete discussion of this bill and of Fulbright's other activities), the avowed purpose of O.T.S. as announced in the Federal Register is to "assist in increasing the technological productivity of the nation by aiding manufacturers and industries in contacting Federal, State and private agencies which can help them in the solution of specific technical problems; by furnishing industry and the public with information on available methods, processes, products, devices, and the like; and by developing and furthering the use of new methods, processes, products and devices." Nothing conservative about the gentlemen who wrote that.



NEW LANDING GEAR—"Superflex" undercarriage developed by the Firestone Rubber Company employs a rubber cylinder filled with compressed air as a shock absorber. In this sketch, the landing gear is extended to full length and the rubber spring is in position to absorb the maximum amount of initial landing impact. The friction material restricts any bouncing tendencies. According to the manufacturer, the only maintenance required is occasional checking of air pressure.

Aid to the Small Businessman—Informed circles in Washington see in this move an extension of the "let's be kind to the small businessman" attitude of the Wallace-Kilgore-Fulbright group. First indication of this trend was the establishment of the "Office of Small Business" in the Commerce Dept. Wallace's original plan was to have Congress pass specific legislation approving such a division, with an Assistant Secretary at its head. Failing to obtain specific Congressional approval, he succeeded in setting up this department in the same manner by a departmental order, with specific approval from Congress for the amount of money to be spent. Thus, if there is any criticism of O.T.S. at some later date, Congress cannot hold itself blameless.

The Office of Technical Services will consist of five divisions; most of them represent reincarnations of dead wartime agencies. The most notable of these divisions is the "Industrial Research and Development" division, which is a rebirth of the old Office of Production Research and Development of W.P.B. The excellent record of O.P.R.D. in sponsoring vital industrial research projects during the war is well known. Washington is wondering however, how such an organization can justify its existence in peacetime. No emergency exists, at least insofar as technical development is concerned; the only reason for existence of such an organization therefore would seem to be furnishing of technical information to those who are not capable of doing their own research or those who cannot afford it. If such be the case, this would seem to be in direct opposition to the well tried independent American method of competitive research and development.

Showing an even greater tendency in this direction is the "Technical Advisory Services" Division of O.T.S. This is the old technical Advisory section of the wartime Smaller War Plants Corporation, which was headed by Maury Maverick. The Smaller War Plants Corporation was the official administration expression of aiding the small businessman during the war, which has been given new life in the Commerce Department program.

Presumably, the Technical Advisory Services Division would furnish gratis to any citizen who asked for it the fruits of the work of the Industrial Research and Development Division.

One million dollars has been allotted by the O.T.S. as an initial expenditure in letting out research and development contracts to university laboratories and industrial research institutes. The nature of the research to be carried out under these contracts will be decided by O.T.S. An "Industrial Research Advisory Council," to consist of from four to six men, chosen from industry and science would offer suggestions to O.T.S. as to what kind of work should be done. In addition, O.T.S. has prepared a guide for the submission of proposals for research by industry. In every case, however, O.T.S. will be the final judge as to what kind of research will be sponsored.

An appraisal of industry's reaction to the program is not yet possible since it has not begun to function. Certain indications are however apparent .The amount of money is approximately one million dollars, to be expended in one year's time for all research done by O.T.S., which, to many industrial circles is extremely significant. This amount is much more than many small businesses could afford to spend for research, even in ten years time. It is however, much less than the amounts allotted for research by many large companies, whose research includes investigations of a basic scientific nature. Thus, the conclusion seems to be that although the facilities of O.T.S. are available to anyone, only small business would make use of this service. Further, since the status of O.T.S. is not assured beyond July 1, 1947, it certainly would not take on itself any research or development work of a basic long range nature. This again would be in favor of the small businessman since all of his problems are immediate ones and his finances do not permit him to wait until his markets are ready for new discoveries. The talk one hears about the Commerce Department on "cartels" and the "potentials of big business" would seem to bear out the contentions of those who claim that the activities of O.T.S. are intended to bolster small business or those businesses which are not enterprising enough to do their own research.

Wartime Residue—Two other divisions of the Office of Technical Services are completing unfinished wartime business of the Department of Commerce and other Government agencies. One of these was the Technical Industrial Intelligence Committee which operated under the joint Chiefs of Staff, with funds provided by the Foreign Economic Administration. The purpose of the T.I.I.C. was to send trained technical investigators into Germany, Japan and other enemy countries to report on the state of enemy technology and to bring back specific examples of advancements in engineering and science, so that American industry could benefit from these discoveries. This committee is now a division of the Office of Technical Services, known as the Technical Industrial Intelligence Division. Although hundreds of investigators have been through Germany and Japan and have submitted reports on their findings, O.T.S. claims that many more investigators are needed to complete the task.

In charge of issuing reports to industry and other interested parties on the findings of the technical investigators in enemy countries and on many American wartime discoveries which are just now being released has been the "Office of the Publica-

tion Board" of the Commerce Department. This has now been incorporated into the O.T.S. under the name of the "Bibliographic and Reference Division." Thus far, reports made possible by this organization have been mostly on foreign developments, but an increasingly greater percentage of domestic reports is being issued. Spokesmen for this organization say that there will be enough foreign material to keep its personnel busy for at least another eighteen months.

These people are also looking ahead and claim that this division could have a legitimate peacetime function of compiling all federal technical reports and articles into one central reference bibliography. Why this should be a function of the Commerce Department, rather than the Library of Congress, is not indicated. Members of this organization claim that the job done so far in issuing foreign technical reports is being overwhelmingly appreciated by industry. They report that several large industrial firms have organized special staffs to do nothing but study these foreign reports for the useful technical information contained therein.

Inventions—Head of the Office of Technical Services is John Green, who was chief of the National Inventors Council. This was a semi-official organization of industry and government men whose function it was to advise inventors on discoveries which effected national security. Mr. Green has degrees in engineering and law and his previous experience included working as a patent office examiner. When the Office of the Publication Board was set up to issue enemy technical reports, Mr. Green was made head of the new organization. Although he is now in charge of the Office of Technical Services, he has not forgotten his original "baby," the National Inventors Council, which has also been included as a division of O.T.S., under the name of "Inventions and Engineering Division." According to the notice in the Federal Register, this Division "is generally responsible for performing those functions of the Office of the Technical Services relating to counseling and assisting inventors; stimulating inventions by indicating areas and cities in which inventions are needed and in general encouraging the development of inventiveness as a natural resource."

Another function of this division is to "provide inventors with information as to the functions and procedures of the patent system; their rights and interests in inventive ideas; research, development and test facilities available from business and private sources; and financing and exploiting their marketing problems; and compiling and making available to inventors information on problems needing solution through new inventions."

What reaction patent attorneys have taken to the new setup is not yet known. Certainly, if the new organization carries out all of its avowed functions, the inventor will have little to do, since all of his problems should be solved by O.T.S. Here again the small inventor or the inventor who does not have enough enterprise to do his own research will be favored over the more established solidly-based inventors.

The personnel involved in this enterprise is listed by O.T.S. as follows: 272 in the United States, 100 American personnel in Europe, 46 German personnel in Europe and 63 industrial consultants working without compensation in the United States. Since these figures include clerical and other non-technical personnel, how much research will actually be accomplished is an unknown quantity.

A.S.M.E. Publishes Metal-Cutting Data Sheets

WITH THE RELEASE of the July 1946 issue of the Metal Cutting Data Sheets, the American Society of Mechanical Engineers makes available to industry information about cutting tools and methods, including new information on the use of carbide tipped tools, culled each month from research laboratory reports and industrial sources.

The new publication is an outgrowth of the metal cutting Data Sheets published during the war by the Society for the

Strain Gages Used to Analyze Stresses in Welded Trusses

TO OBTAIN FUNDAMENTAL research data for the further development of welded H-section trusses of long span and unusual framing. The Austin Company recently concluded a series of tests, using a total of 144 strain gages. The tests were conducted on a pair of 80 foot standard trusses.

H. D. Churchill, professor of engineering mechanics at Case School of Applied Science, directed the tests which were carried on by members of his staff. In the tests, SR-4 strain gages were attached to the joints at both ends of the first diagonal and at certain other important points. They were placed at the same locations on both trusses to provide a check.

The strain gages, made by the Baldwin Locomotive Works, make it possible to measure instantly the average strain in a one-half inch length to one-millionth of an inch. The gage, itself, consists of a tiny loop of copper-nickel wire with a diameter of one-thousandths of an inch—finer than a human hair. This is cemented to the steel surface so that it will change in length exactly as does the steel on which it is mounted. This stretching causes the diameter to decrease and the change in diameter changes the resistance. While this change in resistance is extremely small, it is multiplied by electronic equipment to a point which can be measured by conventional recording devices.

The test loads were applied by placing 60 foot lengths of structural steel on the trusses in a manner which concentrated the full load on purlins located directly above the vertical truss members.

The strain measurements were recorded by three automatic switching units and three SR-4 strain recorders. Each pair of instruments, including a switching unit plus a recorder, made an automatic written record of the strains at 48 gage points, so that 144 individual strain readings could be recorded in about two minutes. More than three miles of wire was required to connect the gages to these instruments.

Testing strain gages during preparation for tests on 80 ft. welded H-section trusses.

Office of Production Research and Development of the War Production Board as an aid to the war effort. The former publication, 10,000 sets of which were distributed free throughout industry, was primarily devoted to milling data obtained as a result of government sponsored research at the California Institute of Technology, Pasadena, California, and the University of Michigan, Ann Arbor, Michigan.

As an A.S.M.E. publication, however, the Metal Cutting Data Sheets have a much broader scope. The Data Sheets are issued in the form of 8½x11 in. punched loose-leaf pages for convenient filing in a folder. Subject matter is presented under ten divisions of the metal-cutting industry. Each subscriber is provided with a set of index cards to facilitate filing. Subscription fee for 12 issues of the Metal-Cutting Data Sheets is $15.00 in the U.S.A. Subscriptions should be addressed to The American Society of Mechanical Engineers, Publication Sales Department, 29 West 39th Street, New York 18, N. Y.

Patent Office Plans

THE UNITED STATES PATENT OFFICE plans to complete its reclassification of all American patents, now numbering about 2,500,000, and is also surveying the need for world wide classification of patents and allied technical literature, according to Casper W. Ooms, Commissioner of Patents.

Speaking at the Symposium of the Division of Chemical Education of the American Chemical Society at Atlantic City, Mr. Ooms called attention to the overwhelming complexity and volume of technical contributions and technical literature now deluging the Patent Office.

In order to carry out the much-needed task of bringing American patent classifications up to date, Mr. Ooms said that at least 100 competent professional persons would have to be employed. At present fewer than 15 persons are engaged in this work, he said, but the Patent Office hopes

Foreign Reports

. . . . . from McGraw-Hill World News

Underwater High-Frequency Tempering of Steel

Underwater tempering of steel surfaces by high-frequency current—now being introduced in Soviet industry on a wide scale—is described in an article in the June issue of Moscow News by Dr. Mikhail Lozinsky, one of the developers of the method. The method consists in immersing both the section to be treated and the inductor in water, which also serves to cool the latter. It is claimed to produce a surface 50 to 100 percent more durable than the previous method.

High-frequency tempering, although widely used throughout Soviet industry, until recently suffered from one shortcoming: The method could not be used to treat the inner surfaces of tiny holes in gears, bushings or narrow slits and intricate grooves. The setup then used simply did not fit into them.

The question of cooling the induction coil, or inductor, inside which the part to be treated is placed, was the heart of the problem. In the original setup this was done by forcing water through the tiny gap—which had to be as small as possible without allowing direct contact—between the inductor and the tempered part. At first, elimination of the cooling system was attempted in order to make the device compact. But the miniature inductors, made of fine copper wire, melted almost immediately.

A magazine article which described underwater repairs to ships' hulls by oxyacetylene welding gave Dr. Lozinsky the idea of heat-treating under water. Water is a poor conductor of electricity and hence the loss of current through it could not be great, and since stem, which was bound to form at the surface being heated, is a poor conductor of heat, the loss of heat would likewise be small. The inductor, on the other hand, would have a perfect cooling system. The first experiments proved that this reasoning was correct.

The underwater inductors are made of either copper wire or narrow strips of sheet copper. They can be made small enough to fit into a matchbox. They are immersed in water one or two millimeters from the parts to be treated, and the current turned on. Apart from the economic effect on industrial productivity, the method is quicker than its forerunner.

Recently the method has been used in heat-treating complicated dies and parts with holes 15 mm. in diameter. The thickness of the tempered surface layer may be adjusted to a fraction of a millimeter.

New Test Method for Alloys

DISCOVERY OF A RAPID SHORTCUT in the formation and testing of alloys is claimed by a Russian engineer, Sergei Vekshinsky, who headed the research department of the Leningrad Electric Lamp Factory during the war.

Instead of studying the properties of alloys in their static shape, a process involving preparation of innumerable samples, Vekshinsky's method permits the researcher to run through a whole range of variations of any combination of metals at once, according to Moscow press reports.

It consists of heating two metals to incandescence in a vacuum, then mixing the streams of particles emanating from them and catching them on a screen for examination. The result is sort of a metal spectrum. Vekshinsky says the center of the strip traced on the screen by the outpouring of particles consists of equal quantities of each of the metals, while on one side one of the metals evenly and gradually increase in quantity and on the other decrease equally steadily.

Beryllium Patent Offered

American industry is being offered patent rights on an Italian secret process for producing marketable beryllium alloys—both copper and aluminum—without reducing the element, the Economic Section of the U. S. Embassy in Rome reveals.

The process is owned by SAPPI (Societa Amonima Processi Privati Industriale) and has been used successfully at a factory built at Chivasso near Turin in 1936. The Germans operated the plant and exported all its output from 1940 until the end of the war. Interested firms should contact the principals through the Embassy Economic Section.

British Nickel-Clad Steel

An economical nickel-clad steel plate for heavy uses in the chemical and food manufacturing industries is described in two booklets issued by Guest, Keen, Baldwin's Iron & Steel Co., Ltd.

The steel is "cladded" on one side with a layer of pure solid nickel, which varies between 5 and 20 percent of the metal thickness of the aggregate plate, by rolling it on at a temperature high enough to insure a sound bond. There is said to be little or no tendency to distortion when using the clad plate at high temperatures.

Adding Steam Saves Motor Fuel

A DEVICE TO ADD STEAM to the fuel mixture to produce the desired degree of humidity has been developed by two Swedish motor technicians, Hans Mandin and Eric Nilsson of Norrkoping. Instead of water injection, this system employs steam generated by exhaust heat, piped through a purifier, and mixed with the gasoline as it leaves the carburetor. The device is simple to install and occupies little space, and produces fuel savings of 10 to 12 percent.

to procure the necessary personnel as quickly as space and budget are provided.

* * *

A PROGRAM designed to streamline and simplify the rules and procedures of the Patent Office, has been announced by Casper W. Ooms, Patent Commissioner. The Patent Office has already adopted a new, simplified application form for use by inventors seeking patents. The form will be put into use in the near future.

In addition, a committee of Patent Office employees has been named to study and evaluate the rules, practices and procedure now in use at the Patent Office with a view to formulating changes and revisions that may appear desirable.

The Patent Office is now deluged with a record number of applications that may number 100,000 this year. Because of a shortage of clerical and other help there is now a backlog of more than 106,000 applications awaiting action.

Carbide Dies Receiving Wider Use

CARBIDE DIES have become standard equipment in the drawing of round wire and for drawing tube, bar stock, and shapes, according to Earle Glen, Carboloy Company, Detroit. In addition, they are now being used for deep drawing of sheet metal. The latter development is relatively recent.

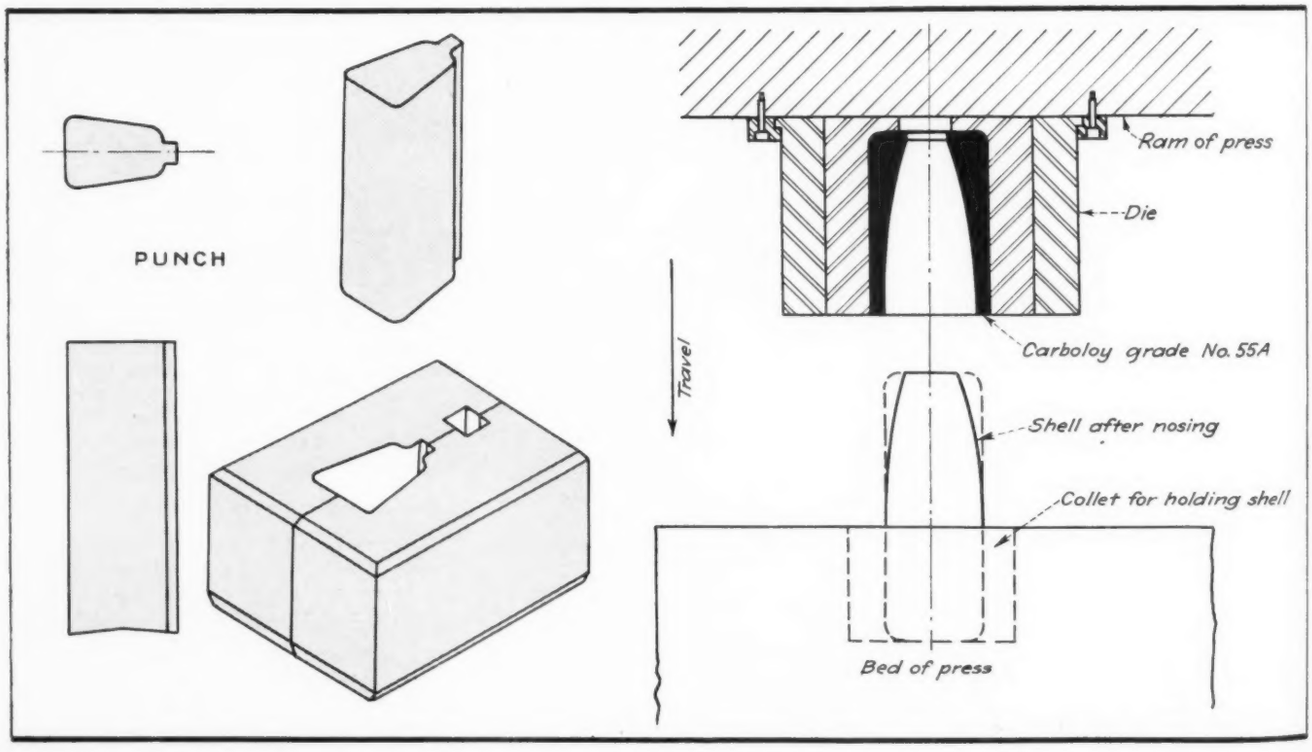
Today, there are in regular service carbide dies for deep drawing worth inside diameters as large as 13½ in. Since a carbide nib of this size weighs more than 100 pounds, special techniques for the production of such large dies have had to be developed. One such is the "hot press" method in which the three major operations of pressing, semi-sintering, and sintering are combined in a single operation, with the final die nib having the same close tolerances and high surface finish as the smaller sheet metal dies.

Major advantages of the carbide dies are: (1) an appreciably longer service life than comparable steel dies; (2) cut down the amount of servicing required; (3) noticeably step up production; and (4) often give a better surface finish to the metal, thereby reducing subsequent buffing time. In addition, with carbide dies the total number of drawing operations is less thus also reducing the total die inventory of a plant. Decreased down time of equipment for maintenance and servicing is another item mentioned.

The life has been increased by a ratio of 20 to 1 over what it was when steel dies were standard. Moreover, whereas steel dies needed repolishing after every 2,500 draws, the carbide dies do not need servicing until after some 10,000 comparable operations. No changes were necessary in press operation lubrication practice, or die design when the change-over from steel to carbide dies was made. Some changes were made in die size however. As a result of these changes, the total number of drawing operations was reduced by some 15 percent. For instance, a cylinder previously requiring six draws now needs but five draws inasmuch as the carbide dies permit heavier drafts to be taken without reducing the service life of the dies materially.

Application of carbide punches and dies to blanking and laminating is steadily increasing. Good performance is being obtained on blanking steels and other metals in thicknesses up to .025 in. in rounds and simple shapes. Development work is being carried on more complicated shapes and thicknesses greater than .025 in. Averages of 1,500,000 to 2,000,000 punches are being obtained with carbide dies and punches between re-grinds, as contrasted with a comparable average of some 50,000 operations for steel punches and dies.

War experience proved the practicality of cold nosing steel with carbide dies. By V-J Day, carbide cold nosing dies were so widely used in the shell industry for projectiles up to 105 mm. that such dies were rapidly becoming "standard." Use of carbide dies for such cold forging work involved no departure in practice, although in most cases slightly more care was exercised in lining up dies.



(Left) These carbide stator laminating dies are made in two halves, clamped together so they can easily be taken apart for regrinding. The punch tapers towards the center, providing freer cutting and prolonging punch life. (Right) The carbide dies used for cold nosing artillery shells up to 105 mm. in size produced from 1,000,000 to 3,000,000 shells per die, as contrasted with 50,000 shells per steel die. The carbide dies did not wear and gall.

RESEARCH NOTES

Government And Industry On Aeronautical Research

A National Aeronautical Research Policy designed to insure maintenance of American air supremacy in the future, was made public recently following a joint meeting of the National Advisory Committee for Aeronautics and the NACA's Industry Consulting Committee.

The Aeronautical Research Policy includes a clear-cut definition of functions of the major technical elements comprising American aviation, with the intent that maximum technical progress in aviation can be achieved through coordinated efforts. Fundamental research is considered the function of the Government's NACA; application of research results in the design and development of improved aircraft and equipment, both civil and military, is the function of the aircraft industry; evaluation of military aircraft and equipment developed by the industry and the exploration of possible military applications of research results, are considered to be the function of the Army and Navy; and expedition of the practical use in civil aeronautics of newly-developed aircraft and equipment, in so far as Government assistance may be necessary, is the function of the C.A.A.

The Aircraft Industry's endorsement of this policy was made by J. H. Kindelberger, President of North American Aviation, Inc., who is chairman of the NACA's Industry Consulting Committee. Other Government aeronautical agencies participating in the formulation of the policy through their membership on the National Advisory Committee for Aeronautics are the Army and Navy air organizations and the Civil Aeronautics Administration.

Army Ordnance Farms Out Research

Army Ordnance Department is continuing, on a large scale, its pre-war and wartime policy of farming out research and developmental jobs to university, industrial and independent laboratories. To date, 18 contracts have been signed for continuance in post-war period ahead work related to the Ordnance program. More contracts will be negotiated when the amount of appropriation for this type of work in the next fiscal year is settled by Congress.

Army and Navy Explore Cosmic Rays

A flying laboratory has been sent into the clouds to study cosmic rays, the Navy's Office of Research and Inventions announced recently. The project will be a joint Army Air Forces-Navy undertaking.

Major General C. E. LeMay, U.S.A., Deputy Chief of Air Staff for Research and Development of the AAF, announced that the AAF has made three B-29 aircraft available for the Navy's experiments in addition to providing flight crews and aircraft maintenance supervisor personnel.

The original AAF participation in this joint mission was to furnish the airplane. Subsequently, it was discovered that the B-29 had enough surplus space, after provision was made for National Geographic equipment, to enable the AAF to plan its own independent research project.

It will be possible to conduct both projects simultaneously because National Geographic is interested only in determining the effects of cosmic rays that bombard the earth, while AAF's concern is with military problems.

The civilian aspects of the flights will be directed by Dr. Lyman J. Briggs, chairman of the research committee of National Geographic, and Dr. W. F. G. Swann of Bartol Foundation.

The B-29's were based at the Naval Ordnance Testing Station, Inyokern, California. Present plans call for four flights at altitudes of 5,000, 15,000, 25,000, and 35,000 feet, with the high altitude flights to be made last to enable AAF to make adequate preparation of scientific instruments, test kits, and data.

In undertaking high altitude cosmic ray measurements, a specially designed magnetic cloud chamber was utilized. The Navy is interested in finding out from these measurements whether cosmic radiation has an effect on the propagation of microwaves, the possible effect upon human beings when subjected to these radiations at high altitudes, and the effect upon electronic guiding systems.

To explore a region of the atmosphere never before investigated, the Navy sent a miniature scientific laboratory 100 miles into the ionosphere on board a V-2 rocket fired at White Sands Proving Ground, Las Cruces, New Mexico on June 27. In the hope of obtaining answers to some of the problems about the universe Navy scientists at the Naval Research Laboratory, Bellevue, D. C., have prepared the miniature laboratory which will be carried in the rocket's "warhead." Scientists stationed at many observation posts will operate a massive array of equipment to track the rocket in its six minute flight and record the data that it radios back to earth. The June 27 project was conducted in collaboration with Army Ordnance Engineers who have charge of the launching and flight of the missile. (See story on rockets in July "Industry and Societies" section).

Hydroelectric Study in India

A hydroelectric testing high tension laboratory is to be established at the Indian Institute of Science to obtain trained personnel and testing facilities for hydroelectric development in India. The laboratory will be able to test high voltage insulations and other electric machinery of 1,000,000 volts capacity.

MEETINGS

October 10-12
American Society of Tool Engineers—Semi-annual meeting, Hotel William Penn, Pittsburgh, Pa.

October 14-19
Electronic Radio and Television Exposition—Grand Central Palace, New York.

October 16-17
Society of Automotive Engineers—National Transportation and Maintenance meeting, Hotel Knickerbocker, Chicago, Ill.

October 16-19
Electrochemical Society—Fall congress, Royal York Hotel, Toronto, Canada.

October 22-23
Conference on Hydraulic Machinery—Sponsored by the Armour Research Foundation and the Graduate School of Illinois Institute of Technology, with the cooperation of the Western Society of Engineers and the local sections of the A. S. C. E., A. S. M. E. and S. A. E.; Hotel Continental, Chicago, Ill.

October 23-26
National Tool and Die Manufacturers Association—Annual meeting, Congress Hotel, Chicago, Ill.

October 25-26
Society of Rheology—Annual meeting, Hotel Pennsylvania, New York.

October 29-November 1
Refrigeration and Air Conditioning Exposition—Cleveland, Ohio.

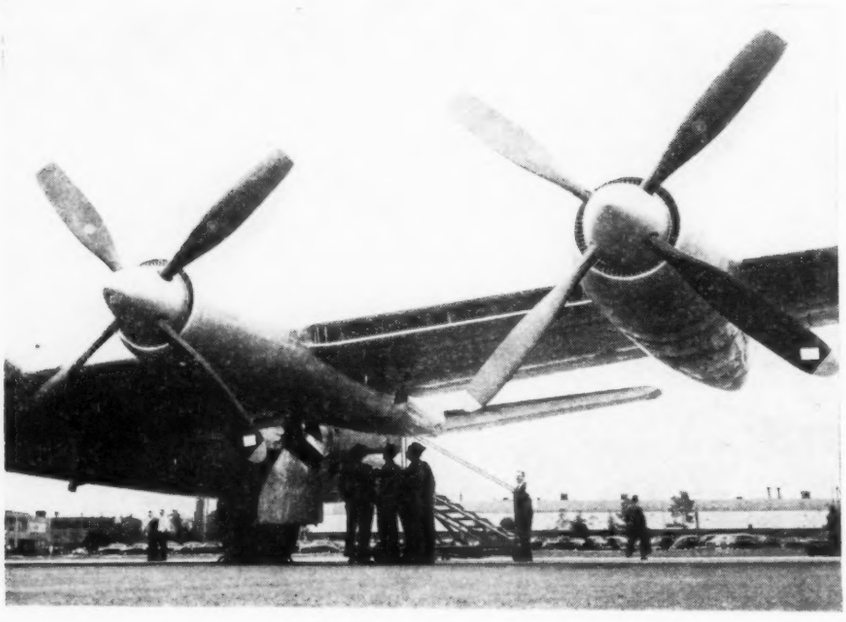
November 4-6
Federation of Paint and Varnish Clubs—Annual Paint Industries' Show, Hotel Claridge, Atlantic City, N. J.

November 7-8
Industrial Management Society—Annual Time and Motion Study Clinic, Continental Hotel, Chicago, Illinois.

November 15-24
National Aircraft Shows—Cleveland Airport, Cleveland, Ohio.

November 18-22
American Society for Metals—National Metal Exposition and annual meeting, Municipal Auditorium, Atlantic City, N. J. Held in conjunction with the Exposition at Atlantic City will be the annual meetings of the **Iron and Steel Division**; the **Institute of Mining and Metallurgical Engineers; The American Welding Society** and the **American Industrial Radium and X-Ray Society.**

December 9-11
Society for Experimental Stress Analysis—Annual meeting, including a Symposium of Telemetering of Aircraft Flight Observations, Hotel New Yorker, New York, N. Y.



"Rainbow" Marks High in Large Plane Speeds

REPUBLIC AVIATION CORPORATION'S military photo reconnaissance plane, the XF-12, was revealed recently as the fastest longe range, four-engine transport-type airplane in the world. The military prototype of the 46-passenger "Rainbow" transport plane, the XF-12 is said to mark a new era for multiple-engined passenger airlines in global transportation. Its speed is over 450 miles per hour and its service ceiling is over 44,000 feet.

The 51-ton plane, propelled by more than 12,000 horsepower and measuring 129 feet, two inches from wing tip to wing tip, was developed as a flying photographic laboratory for the Air Technical Service Command of the Army Air Forces.

The plane is powered by four 28-cylinder, 3,000-horsepower Pratt and Whitney engines turning 16-feet, 2-inch four-blade propellers. Engineers expect the plane to operate at very high altitudes at speeds comparable to wartime fighter planes.

Slightly smaller than the B-29, the cigar-shaped fuselage of the new plane is 93 feet, 10 inches long. Height to the top of the stabilizer is 28 feet, 4 inches. Unique in its design for specialized reconnaissance duties, the XF-12 carries complete radio and radar equipment, armor plate and flash bombs for night photography. Specifications include three camera stations—one split vertical, one trimetrogron, plus one vertical camera station. Darkroom equipment for loading film is included.

Housed in nacelles the size of P-47 fuselages, the four engines have 2-speed cooling fans. Each engine is fitted with two General Electric superchargers. The long nacelles are tapered over exhaust turbines from which engineers hope to garner additional thrust of an estimated 100 horsepower per engine. Air intake of the plane is in the leading edge of the wings. It has heat de-icing of wings and tail.

Republic recently announced that, with changes in power plant and interior design, the XF-12 military version is actually the forerunner of the 46 passenger Republic Rainbow Transport. This commercial version, according to Republic, will be the fastest means of long range aerial transportation ever afforded mankind. Scheduled for delivery to the airlines in 1947, the Rainbow will be driven by more powerful engines of later Pratt & Whitney design developing even greater horsepower, and fuselage of the passenger liner will be approximately five feet longer than the XF-12.

Plastic Tubing for Aircraft Limited by Temperature

SUBSTITUTION of thermoplastic for metal tubing in aircraft is seriously limited by the effect of temperature variations on thermoplastics, according to a U. S. Army Air Corps technical report released by the Office of the Publication Board, Department of Commerce.

The report presents the results of wartime tests on various types of semiflexible and rigid thermoplastic tubing over a temperature range of 158 to 67 deg. F. Cellulose acetate-butyrate, vinylidene chloride, nylon, ethyl cellulose, methyl methacrylate, copolymer vinyl chloride acetate, and other plastic materials were tested.

Tests of the tubings were made over such a wide temperature range because the properties of the various thermoplastics were found to vary greatly with temperature changes. The bursting pressure for one type of material at 158 deg. F. for example, may be only five percent of its bursting pressure at 77 deg. Static strength increases with lowered temperatures, while impact strength and elongation decrease, according to the report.

Cellulose acetate butyrate and vinylidene chloride showed promise of being satisfactory all-around tubings. However, actual service tests of these materials had not been made when the report was written.

Resistance Welding Movie Announced by General Electric

A 16MM. ALL-COLOR sound movie, "This Is Resistance Welding," designed to promote an understanding of the operating principles and advantages of resistance welding, has been announced by the Gen-

eral Electric Company. The new film, which requires about 30 minutes to exhibit, is available for loan without charge through electric utilities, welder manufacturers, and G. E. offices throughout the country for exhibition before engineering and manufacturing groups, schools, and other groups interested in resistance welding. Operating principles and capabilities of the four principle types of resistance welding—spot, seam, projection, and flash welding are described in detail.

DISCUSSIONS AND COMMENTS FROM READERS

"Poor Recognition Breeds Frustration"

To the Editor:

I was much interested in your editorial on the mishandling of engineering personnel. What you had to say was much to the point and I feel that an unsympathetic attitude on the part of management toward engineering is all too prevalent in this country.

May I bring to your attention the lack of sympathy toward new ideas, which are engineers' "brainchildren." Too often projects are carefully prepared and the request for funds goes to the front office, never to be seen again. If it comes back at all, it is just with some remark to the effect that the president did not like it. No reason or tactful explanations are given. Management should realize that it would almost pay to let the engineers build a new model once in a while, however, unproductive it seems, just to keep them on their toes. —B.E.P.

"The Suppression of Patents"

To the Editor:

When I was Executive Director of the National Patent Planning Commission, I made a very careful investigation of the factors which justify the non-use of patents and of the various charges of suppression which have been given general publicity.

I am convinced that American industry is not guilty of suppression of patents to any significant degree. Non-use does not necessarily involve the suppression of a patent. Patents are often shelved pending financial, commercial, or technical development. A patented article or process for which there is a substantial public demand will not be kept from the market very long. —A. A. Potter
Dean of Engineering
Purdue University

Functions of Patent Attorney And Inventor Discussed

To the Editor:

I read with considerable interest the article "How to Disclose Your Invention for Maximum Protection" by William K. Rieber, in the May issue of Product Engineering. His article is extremely well written and covers a very important subject.

From my own experience with inventors, attorneys and patents, however, I can hardly agree with some of Mr. Rieber's proposals.

In regard to the function of the attorney, it appears necessary, except under ideal conditions, for him to go a great deal further into the technical aspects of the invention than Mr. Rieber suggests, in order to write a comprehensive and strong patent application. I believe he gives Mr. Average Inventor too much credit, when he expects the inventor, in his preliminary disclosure, to cover the art in which the invention falls, a discussion of the prior art, the objects of the invention, and a detailed description of the invention.

There are undoubtedly a large number of inventors who have a sufficient grasp of the subject matter, the necessary leisure time and adequate control of the English language to write a satisfactory preliminary disclosure; a substantial percentage merely produce some sketches or a model without drawings, and say: "Here it is, prepare an application."

It is definitely necessary for the attorney, if he is to do a thorough job, to go a good deal further than merely paraphrasing the inventor's description, and drawing up a set of claims, which is essentially what Mr. Rieber seems to propose. The attorney should see that the drawings and description are as thorough and extensive as possible, and that the claims are carefully and comprehensively drawn, with as few loopholes as possible.

Often the inventor, even if he prepares and reads the description, is not in a position to go far enough to cover the necessary ground.

A thorough reading and study of the specifications and claims by the inventor, before filing, cannot be overemphasized. In a number of patents that the writer has analyzed, a few mischosen words, which might have been caught by the inventor, have made patents representing substantial investments practically worthless, if they were ever litigated.

In the analysis of thousands of patents from a manufacturer's standpoint, repeated minor flaws in claim drafting and the extreme tight wrapping of claims have only too often resulted in patents which are of little commercial value, let alone capable of standing up in a barrage of litigation.

Mr. Rieber's reference to the ostrich-like attitude on the part of inventors is extremely well-taken. While specific figures are difficult to obtain, too large a percentage of inventors indicate by their inventions and by their attempts at commercialization, that they are oblivious to the state of the field in which they are working, and that they often delude themselves as to the commercial possibilities of their invention.

It should be the function of the patent agent or attorney to guide the inventor in all phases of patent problems and not merely confine himself to the legal aspects of the invention and correspondence with the patent office. While this may require a considerably higher order of training and probably a greater degree of specialization than has been prevalent among patent attorneys in the past, the signs seem to point definitely in that direction.

After all, the drawings and specifications, as filed, remain substantially unchanged throughout the prosecution. It is, therefore, at the time of the initial filing of the application, that much of the ultimate damage is done.

Please accept this not as criticism, but rather an attempt at defining the scope and function of the patent attorney, which should, in my opinion, be considerably broader than the author indicated.
—S. Bertrand Barnard

To the Editor:

Mr. Barnard's letter regarding my last article in Product Engineering is much appreciated. His points on the duties of a patent attorney are well taken. However, in expositions of this sort it is usually best to deal with one general subject at a time. The subject matter in this article was directed to the attention of inventors. I agree that the attorney should have a good grasp of the technical aspects of the art involved; but he should not have to do the inventing for the client.

My work, which is principally the preparation and prosecution of applications, has too frequently brought me in contact with alleged inventors who feel that mere recognition of a problem is invention, leaving the solution up to the patent solicitor. This is a practice in which the Patent Office and the courts very rightly take an extremely dim view.

Mr. Barnard is absolutely right when he states that a good patent attorney should guide his client in all phases of patent problems, but a good inventor should be well versed in all phases of invention—including disclosure. —William K. Rieber

Errors Reported in Fine Pitch Gear Standards

To the Editor:

The A. G. M. A. Standard on Gear Inspection, No. 236.01, which you published in your August issue, is a fine job. I discovered two typographical errors, however and would appreciate your publishing these corrections:

On page 136, in the table for item 172, the total composite error for Class 1 should be 0.003 instead of 0.0013.

On page 136, in the formulas explaining Data Sheets 1, 1A, 2 and 2A, item should read $C_t = KC$ instead of $C_t = kC$.

—Louis D. Martin
Gear Engineer
Eastman Kodak Company

Theory and Design Of Steam-System Vent Valves

FERDINAND JEHLE
Hoffman Specialty Company

Methods and principles underlying the selection and design of vent valves. Radiator vent valves for atmospheric and vacuum systems are discussed; results of comparative tests of valves are given; typical design calculations are included.

A VENT VALVE functions to permit the air to be pushed out of the spaces in the mains and the heating units by the steam and to shut off tightly when the steam reaches it. It must also prevent the escape of water, if water is present, by momentary closing until the water has drained. Special vent valves are used on steam systems that operate below atmosphere pressure. These valves are usually known as vacuum valves. They have the added function of closing at pressures below atmospheric.

To perform these various functions properly, design of vent valves must provide:

First, some thermostatic means, usually contained in a float, that will cause the valve to remain open while relatively cold air is forced out; close when the higher temperature steam or vapor reaches it; and reopen upon lower ambient temperature.

Second, means for guarding against the passage of water so that the vent closes when water reaches it. Furthermore, the water must be discharged promptly. As soon as the water recedes, the vent must open again.

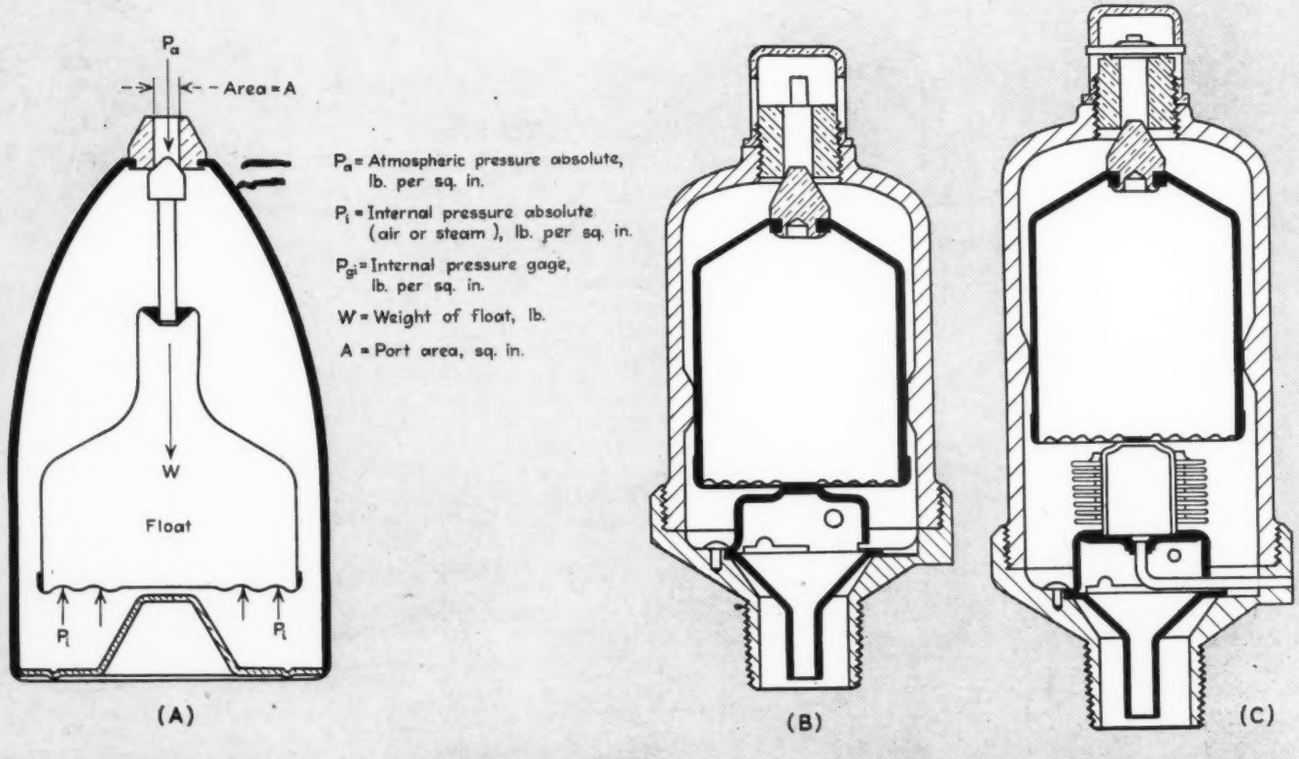
Third, venting area of sufficient size to free the system of air in a reasonable length of time and to reopen, after having closed to prevent the escape of steam or water, without too great a pressure drop. An adjustable venting rate is desirable, particularly on large systems.

Fourth, seals good enough to hold relatively high vacuums for long periods, where installations are made on systems with pressures below atmosphere.

THERMOSTATIC ELEMENT. Practically all air valves today use the float as a thermostatic element. The float bottom, or diaphragm, Fig. 1, expands when heated thereby closing the vent to the passage of steam by pressing the needle against the valve seat. The float rises when the valve fills with water, thus closing the vent and preventing the passage of water.

To make the diaphragm respond to temperature, a small quantity of volatile alcohol and water is put into the float. The float is then exhausted of air and sealed. When such a float is heated, the volatile liquid vaporizes and the pressure created expands the diaphragm. The proportion of alcohol and water is governed by the stiffness of the float diaphragm. If the diaphragm is quite stiff, a higher percentage of alcohol is used. If it is more flexible, a lower percentage of alcohol is required. The percentage of alcohol in

Fig. 1—(A) Typical radiator vent valve showing forces acting on float; (B) Main vent valve (open) showing patented funnel design for water drainage; (C) Main vent valve having bellows construction for large travel in vacuum systems.



meeting the stiffness requirements may vary from almost no alcohol to all alcohol. From each run of the press making the float diaphragm, samples are taken and the exact filling for the entire run is determined by test.

PREVENTION OF WATER PASSAGE. The float obviously must be lighter than water, else it would not rise to force the needle against the seat when water fills the valve. Good design for quick shut off requires that the line of floatation be kept as low as possible.

Not only must the float shut the vent against the passage of water, it must reopen the vent as soon as the water has receded. This means that the valve must drain itself of water. For different types of valves, different devices are used for draining the water. Particularly during the heating up period when the radiator is relatively cold, large quantities of water are condensed and may surge into the end section of the radiator and into the vent valve.

In the old type radiator, drainage was not such a great problem, for vent valves could be equipped with siphons of considerable length. In the newer radiators the tubes are so narrow that a valve with a long siphon cannot be inserted. Modern radiators, therefore, made the development of the tongue type valve necessary. The problem is further complicated by the fact that the bosses on the radiators into which the valves must be screwed, vary in length. A survey showed that the shortest boss in any radiator is only 3/16 in. long and the longest, 1/2 in. In order for a tongue type valve to drain properly, therefore, something must be done in the design so that the drainage device will adapt itself to this difference in boss length.

One design is shown in Fig. 2. The tongue is hinged and will at all times be close enough to the radiator boss to permit the formation of a drop of water and thus start drainage. The illustration shows where drainage takes place, that is, underneath the tongue. Air re-enters the valve by the space above the tongue. For every drop of water drained, a proportional amount of air enters the valve until enough water has been drained for the float to drop and open the port, thus re-establishing venting. Some radiators, especially convectors, take a straight stem valve and drainage is established in the manner shown for main vent valves.

VENTING RATE. The true venting rate of a valve, which should be as high as possible, depends upon the weight of the float, the size of the vent port, and the maximum pressure at which the valve will reopen. This pressure is usually known as the drop-away pressure. Fig. 1(A) shows the forces acting on the float when the water has drained. Calculations for determining the drop-away pressure are simple. The net atmospheric force is P_aA and the net internal upward force is P_iA. For equilibrium, the upward forces must equal the downward forces so that

$$P_iA = P_aA + W \qquad (1)$$

Solving for W

$$W = (P_i - P_a)A = P_{gi}A \qquad (2)$$

The float will drop from the seat when W is greater than $P_{gi}A$. From this it is seen that a float should be as heavy as possible. On the other hand the float must be light enough to still float. Rather than show a chart between the relationship of drop-away pressure, port diameter and float weight, it was thought to be more desirable to take into consideration also float buoyancy, and substitute float diameter for float weight.

The chart in Fig. 3 gives the float diameter in inches for different drop-away pressures when operating with various port diameters. It is based on correct float weights and upon correct water displacement for buoyancy. If a main vent for example has a 1/2 in. port and is to be used for operation at 5 lb. per sq. in., a float diameter of approximately 6-1/2 in. would be required.

The vent valve must be large enough to let approaching steam push out the air at a reasonable rate. The larger the port in the valve, the quicker its rate, but the ratios shown in Fig. 3 must be maintained. No matter how large the vent port is, if the float cannot drop away to open the port, the rate will be zero. The requirements of a good vent valve therefore include not only a port of ample size but also of proportions that will permit

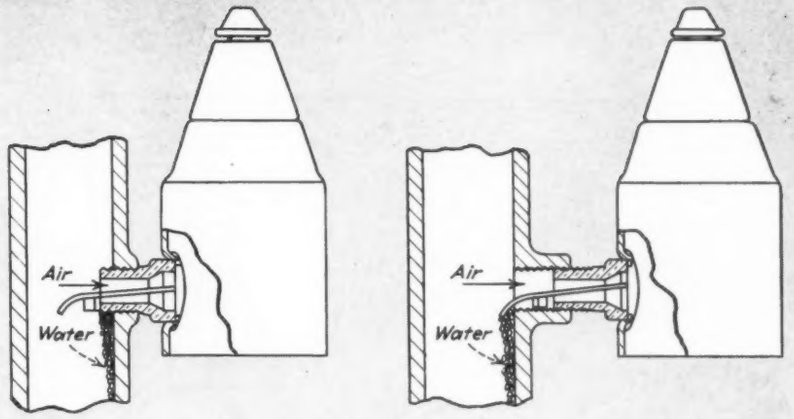


Fig. 2 — Drainage starts underneath the tongue in this device regardless of the position of the valve in the radiator.

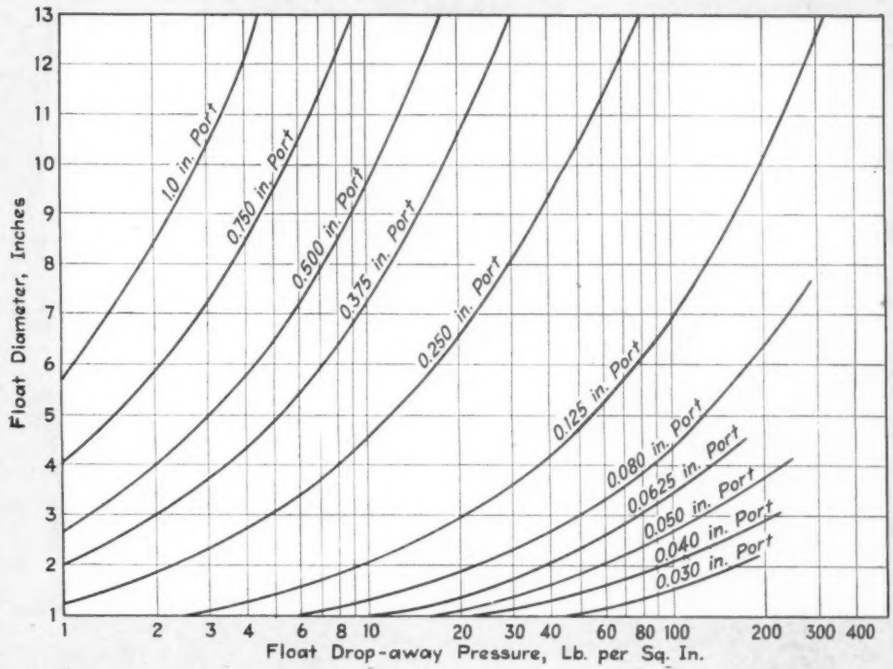

Float Diameter, Inches
Float Drop-away Pressure, Lb. per Sq. In.
1.0 in. Port
0.750 in. Port
0.500 in. Port
0.375 in. Port
0.250 in. Port
0.125 in. Port
0.080 in. Port
0.0625 in. Port
0.050 in. Port
0.040 in. Port
0.030 in. Port
1 2 3 4 6 8 10 20 30 40 60 80 100 200 300 400
1 2 3 4 5 6 7 8 9 10 11 12 13

Fig. 3 — Relationship of port diameter, float diameter, and drop-away pressure.

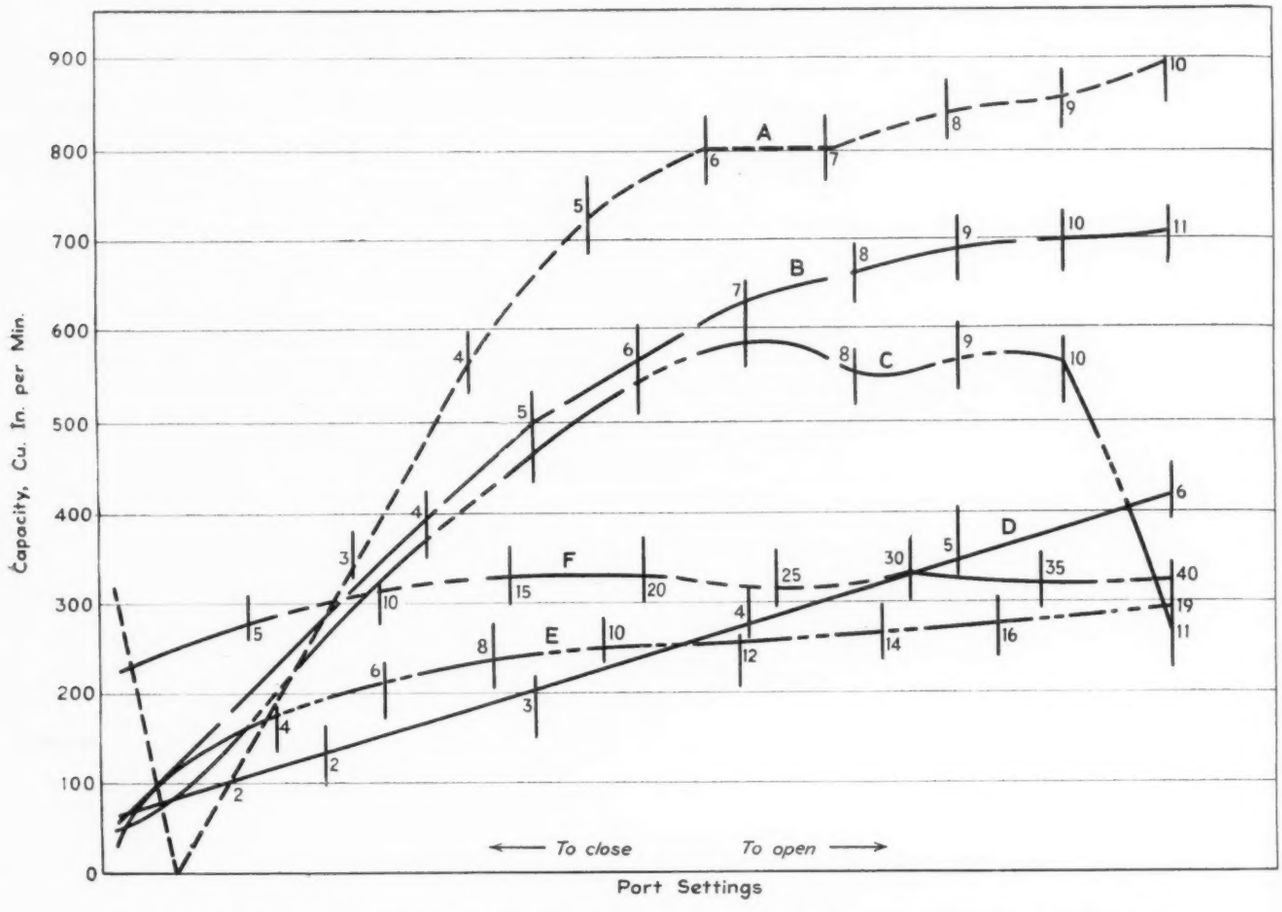


Fig. 4—Results of tests comparing proportional venting rates of six vent valves of different designs.

venting to be resumed without necessitating a drop of pressure in the system.

VACUUM FEATURES. If the valve is to be used on a vacuum system, its design must be such that the vacuum can be formed and held. These two functions are separate and distinct in some makes of valves. Many valves have either a diaphragm or a bellows that pushes up the float assembly, including the needle, by the external atmospheric pressure when the interior of the valve is at pressures below atmospheric. A considerable vacuum is required to operate either a diaphragm or a bellows and the formation of the vacuum is slow.

A check valve is used sometimes in place of a bellows operated needle, but unfortunately a small check valve is not good at holding vacuums. A combination is desirable. The check holds the vacuum during its formation and as soon as the vacuum is high enough, the diaphragm takes over.

Vent Capacity

Some vent valves are used on heating surfaces like radiators, convectors, unit heaters and the like. Others are used on mains supplying these surfaces. They differ in detail only.

If port size is within reason at all, then boiler capacity and not venting capacity of the vent valve determines the heating-up time. A simple example will demonstrate this. Suppose the system has 800 sq. ft. of radiation, divided among 20 radiators and supplied by 150 ft. of 3 in. steam main. Twenty-three cu. in. per sq. ft. of radiating surface in a radiator is a reasonable figure for use in this illustration, which means that the air in the radiators amounts to 18,400 cu. in. The air in the main amounts to 13,345 cu.in. giving a total of 31,745 cu.in. to be removed before the entire system can be filled with steam.

The total heat to be supplied by the boiler during heating consists of raising the temperature of the iron from 70 to 215 deg. F. plus the heat given off to the room by the radiators during this time. Assuming a weight of 5.5 lb. per sq. ft., then 800 sq. ft. of radiation will weigh 4,400 lb., the 3 in. main would weigh about 1,140 lb., making a total of 5,540 lb. Raising the temperature of this iron from 70 to 215 deg. F., assuming a specific heat of iron 0.12, would require 96,396 B.t.u.. Since the radiator is 70 deg. F. at the beginning of the run and 215 deg. F. at the end, it can be assumed for this purpose that only half as much heat will be given off as if the radiator had been up to temperature during the entire period, or about 1,600 B.t.u. per minute.

If "X" min. are required to bring the radiator up to temperature, then the total amount supplied by the boiler for the heat emitted to the room is 1,600 X B. t. u. and the total heat supplied by the boiler for heating the iron, plus heating the room, is 96,396 plus 1,600 X B. t. u. It can be assumed that the boiler has a capacity 50 percent greater than that actually required for supplying the heating load, which would be equal to 1.5 (800 × 240), or a total of 288,800 B. t. u. per hr., or 4,800 B. t. u. per min; hence

$$X = \frac{96{,}396 + 1{,}600\ X}{4{,}800} \qquad (3)$$

Solving for X gives

$$X = \frac{96{,}396}{3{,}200} = 30 \text{ min. (approx.)} \qquad (4)$$

Since the total amount of air to be

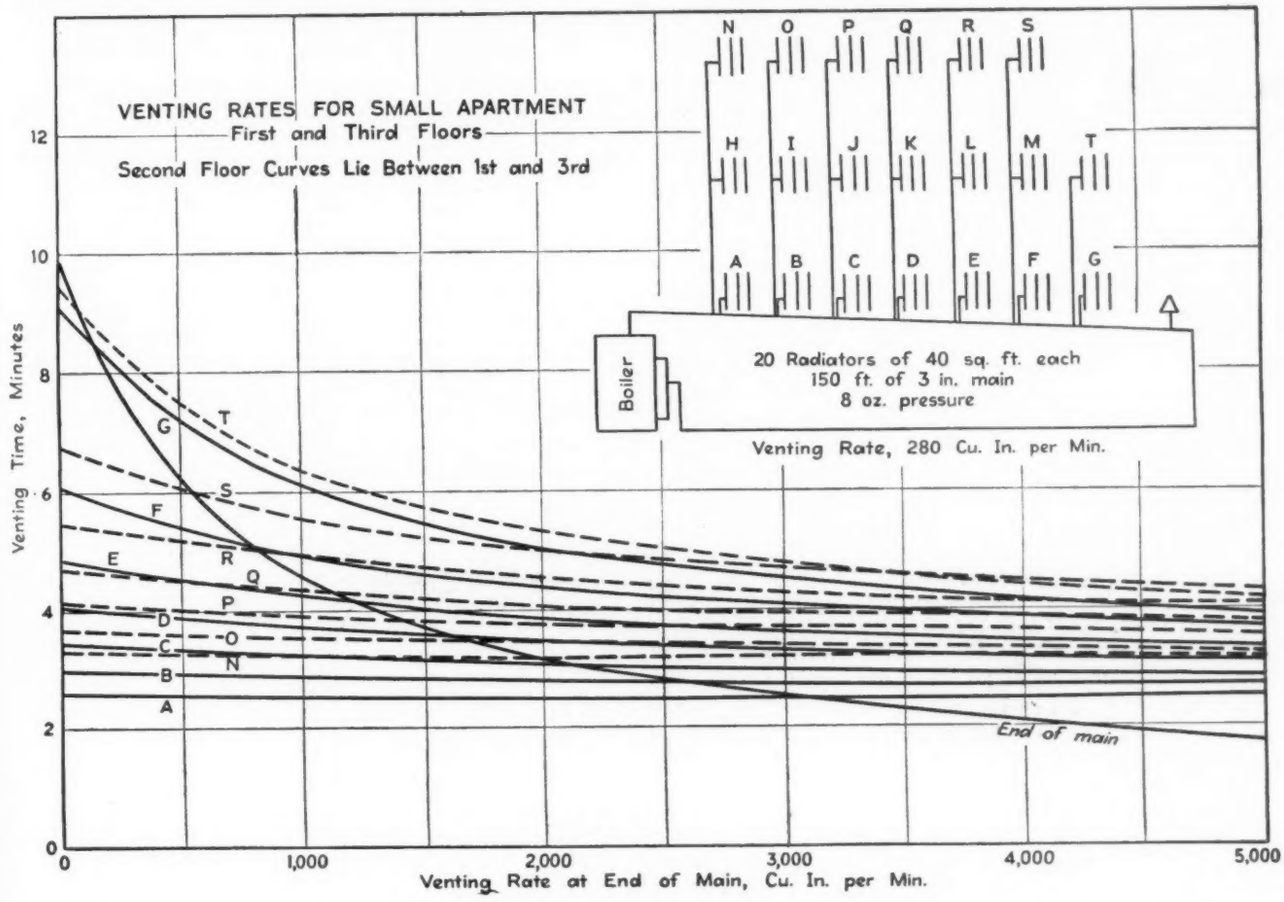


Fig. 5—Rate curves such as these can be used in designing and selecting the capacities of main vent valves.

vented in this installation is 31,745 cu.in. divided among 20 radiators, each of which is supplied with an air valve, each valve would have to have a capacity of approximately 53 cu.in. a min., which is not a great deal for a valve to handle, even at an average pressure of 8 oz. per sq. in. This time would be further reduced if the system is supplied with a main vent on each main in accord with good practice.

Proportional Venting

In a complicated piping system the same pressure cannot exist at each radiator, particularly during the heating-up period when the flow is large. Different venting rates are desirable for the various radiators so that, regardless of size or distance from the boiler, all radiators heat up in approximately the same time. To do this, the smaller radiators must vent at a slower rate than the larger ones, and those closer to the boiler must vent at a slower rate than those farther removed. Usually a manufacturer makes two kinds of vent valves, those which have fixed ports and those which have adjustable ports for varying the venting rates. The former is made chiefly to meet competition. It functions quite well on systems where the pressure is relatively high. The latter functions better on the large systems operating at comparatively low pressures. Good design requires that adjustment of these rates be easy of accomplishment by the user and that they be proportional. Not all vent valves on the market fulfill these requirements. The chart in Fig. 4, shows the results obtained on various valves bought in the open market and tested at 8 oz. pressure.

Valve *A* on this chart has 10 adjustments. The smallest adjustment, which should give the smallest discharge of air, gives a flow of approximately 300 cu. in. per min. Between settings 1 and 2, the valve shuts off completely. Between setting 2 and 6 the flow increased reasonably according to a straight line. At setting 6 the flow is 800 cu. in. per min. The flow increased barely 100 cu. in. per min. between settings 6 and 10. This valve has, therefore, three faults; first, it can be shut off entirely; second, the lowest setting does not give the smallest discharge; and third, beyond setting 6 there is not enough gained to be worth while.

Valve *B* has 11 adjustments and operates well up to the seventh setting. From there on the gain is not worth while. Valve *C* operates well until the seventh setting is reached, then discharge is practically a straight line to the tenth. At the eleventh where the discharge should be the largest, it is only half as much as at the seventh.

Valve *D* has only 6 settings but these are pretty evenly divided and all follow on a straight line. Valve *E* has 19 settings. The behavior of valve *F* is peculiar. It has 40 settings but beyond the 10th its delivery is almost constant.

It would seem from studying the chart that valve *D* is the most desirable. Its capacity is only a little over 400 cu.in. per min., which is sufficient, but the rates at the different port settings lie in a sloping line, which is of great importance, and incidently, its drop-away pressure is reasonable, about 3-1/2 lb. per sq. inch.

Valves for Mains

Vent valves on steam mains are always desirable although there are many small, single pipe heating plants that, in the

interest of saving a few dollars, are not so equipped. Vent valves for mains are designed for conditions of operation somewhat different from those for radiators.

In a main vent it is not necessary that the float drop-away pressure be equal to the maximum steam pressure used. The reason for this is that the system seldom fills with air down to the main vents, and then only when the pressure in the boiler is rather low. In other words, the main vent is called upon for work usually only during the heating-up period.

For a main vent on a zone controlled by a gate or globe valve, the condition may be a bit different. If the zone valve is spun open, the vent may be subjected to boiler pressure immediately and before the air is all expelled. It may, therefore, suck shut or fail to open if closed on flotation. In an installation of this description, the use of two or more valves of smaller port diameter instead of one with a large port is good practice.

The procedure in designing a main vent valve will be illustrated by calculations for a heating system for a small apartment. Suppose the system has a total radiator surface of 800 sq. ft. in 20 radiators of 40 sq. ft. each; on three floors; tubular radiation having a volumn of 15 cu. in. per sq. ft.; 150 ft. of 3-in. main; venting rate of the radiator vents of 280 cu. in. per min. and venting pressure of 8 oz. per sq. inch.

The venting rate is assumed equal for all radiators, which of course is not exactly true. If a different rate is assumed for each radiator, a mathematical problem, not justifiable in this case, is encountered. The error is on the correct side. Fig. 5 shows a schematic arrangement of the radiators and piping connections to the boiler.

The chart, Fig. 5, which is based on the venting capacity of the main valve and the venting time of both main vent and radiator vents, makes possible a graphical solution. Curves representing the venting of radiators on the second floor, with the exception of that of radiator *T*, are not plotted. All of them lie between limiting curves of radiator *T* and of those on the first and third floors. It is interesting to note that the venting rates of all the radiators approach parallelism. They would never meet. In other words, it is impossible, regardless of size of main vent, to have all of the radiators filled with steam at exactly the same time.

Without the main vents, the radiator at *T*, the farthest from the boiler, vents in 9.5 min. and *A*, the closest vents in 2.5 min. making a difference of 7 min. A main vent of 2,000 cu. in. per min. appears ample. Using one of this capacity, the following results are obtained: *T* now vents in 5.5 min. and *A* in 2.5 min. with a difference of 3 min. The heating times of all the radiators will be more nearly equalized by the use of a main vent.

Unit heaters are frequently installed in apartment houses to provide individual control over the heating facilities and in other places where no central heating system is available.

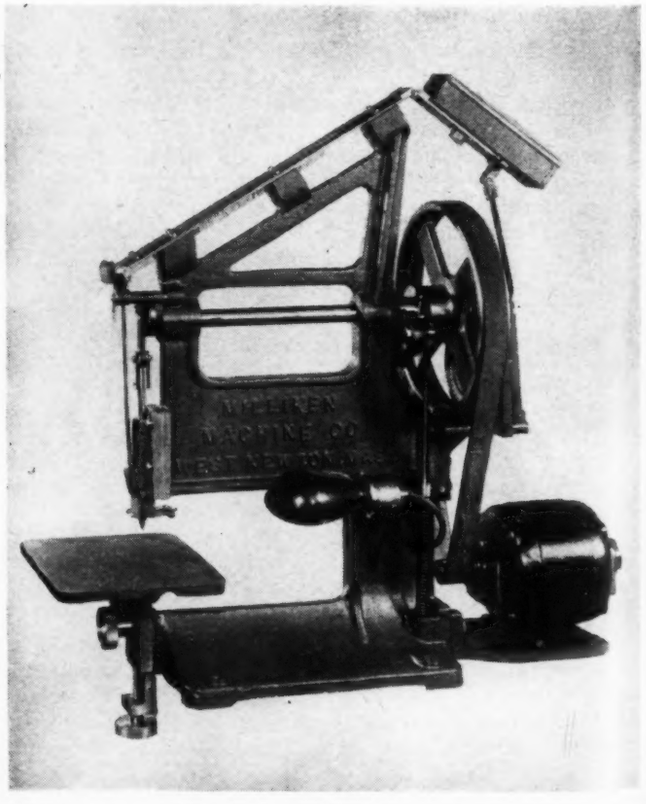
This kind of heating surface presents a venting problem that to some extent is different from venting ordinary radiators. The chief difference is the smaller air content of the unit heater. Since the volume is small the vent valve must be able to vent a small amount of air at frequent intervals if the heating surface is to be kept free of air. To make matters somewhat more difficult, a unit heater usually works on relatively high pressure.

Supposing that a unit heater operating at 20 lb. per sq. in. is equipped with a vent valve having a float diameter of 1-7/8 in. If this valve has a port diameter of 0.250 in. it cannot open until the pressure in the heater has dropped to about 2 lb. per sq. in. If the same valve has a port diameter of 0.125 in. the pressure would have to drop to about 9 lb. per sq. in. before venting could be resumed. With a port diameter of only 0.080 in. no pressure drop whatever would be necessary to re-establish venting. Expressed in another way: The valve with either the 0.250 in. port, or the 0.125 in. port would permit the heater to get cold, but with the 0.080 in. port the heater would stay hot at all times. A vent with approximately a 0.050 in. port and a float diameter of 1-7/8 in. is therefore ample to vent a unit heater and keep it hot up to 35 lb. per sq. in. with safety.

Precision Nailing Machine Eliminates Hand Assembly

DESIGNERS who have avoided precision nailed, pinned or riveted construction because of the high cost of hand operations will find a solution to their problem in this precision nailing machine. Developed by the Milliken Machine Company, Hastings Sales Engineering Div., Boston, Mass., the machine is claimed to have increased production over hand methods by 200 percent on such operations as setting the hinge pins of sun glasses, attaching pigtails to carbon brushes, and assembling plastic products. The unit is sufficiently light to allow mounting on a portable work bench, and is driven by a 1/4 hp. motor. The manufacturer modifies the machine to the specific nail headsize required by the user.

Nails can be angled, countersunk or partially inserted. The sustained driving action, as opposed to the usual blow of larger units, allows a nail to pierce metal and plastic sheets without predrilling. Thin strips of hardwood can also be handled without splitting. By means of jigs holes can be located for riveting such items as bearing assemblies.



Sum and Difference Method For Calculating Stresses in Disks

A. M. G. MOODY and M. L. LAND
Engineering Department, Elliott Company

Standard method for calculating stresses in rotating disks that can be readily adapted to wheels having any type of profile that can be closely represented by a series of parallel sided concentric disks. The method is presented in tabular form with a description of ways to simplify the calculations.

MANY METHODS have been developed for calculating elastic stresses in rotating disks. Of these the "Sum and Difference" method has been adopted as standard by the engineering departments of most companies building steam turbines and similar rotating machinery. There are good reasons for this preference. The method is simple and flexible, it can be learned readily and applied to a wide variety of problems.

Most methods have some special characteristic that gives them an advantage in dealing with a particular type of wheel. Special equations can be developed for analyzing wheels of hyperbolic profile, but for wheels having other types of profile, these equations might not be at all suitable. Consequently, rather than have a separate method for each type of wheel, it is desirable to adhere to one that can be adapted readily to any type.

Many engineers are not aware that a large amount of work has been done in this field, and especially of the wide utility of the Sum and Difference (or S and D) method. An exhaustive critical study of disk stresses, with a comparison of a number of methods of approach, has been made by S. Tumarkin ("Methods of Stress Calculation in Rotating Disks," Central Aero-Hydronamical Institute, Russia. Translated in N.A.C.A. *Tech. Memo.* No. 1064.) The Sum and Difference method has been excellently presented by H. Haerle ("Strength of Rotating Disks," *Engineering*, Aug. 9, 1918).

The Sum and Difference method of stress calculation employs a standard set of curves for stresses in a disk of uniform thickness, that is, a parallel sided disk. These curves comprise two families, the S curves representing the sum of tangential T and radial R stresses throughout the disk, and the D curves representing the difference of these stresses, thus

$$S = T + R \quad (1)$$

$$D = T - R \quad (2)$$

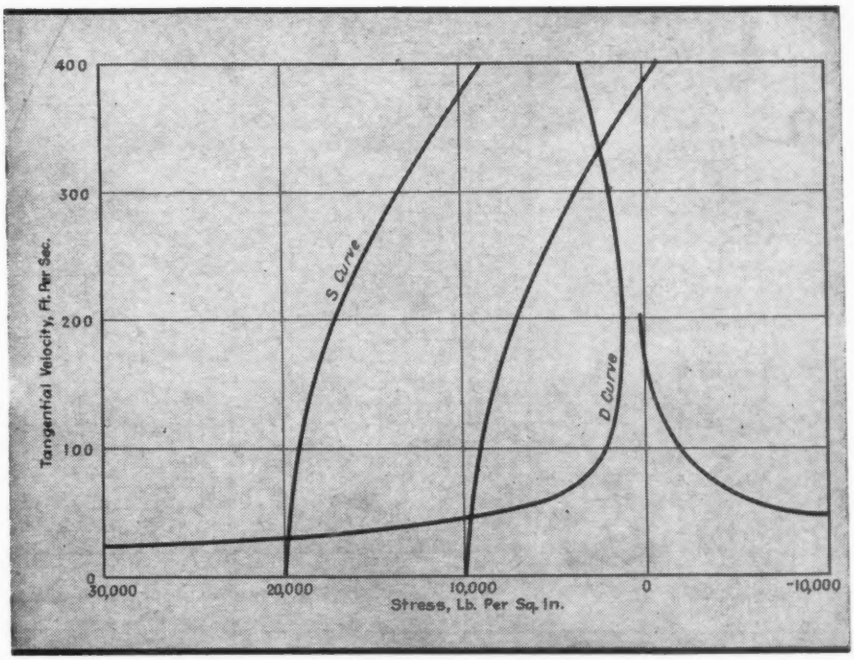


Fig. 1—Typical sum and difference curves.

The S and D curves are plotted from equations derived from Stodola's equations (See A. Stodola, "Steam and Gas Turbines") for tangential and radial stresses in disks.

ν = Poisson's ratio
γ = weight of material in disk, lb. per cu. in.
g = gravity constant, 386.4 in. per sec. per sec.
U = tangential velocity, in. per sec.

$$S = \frac{\gamma\,(1+\nu)\,(-U^2+K_1)}{2\,g} \quad (3)$$

$$D = \frac{\gamma\,(1-\nu)\,(U^2+K_2\,U^{-2})}{4\,g} \quad (4)$$

The factors K_1 and K_2 depend upon the physical properties of the disk material and stress conditions at the bore and rim. For a disk rotating at a given speed, K_1 and K_2 are the only variables for any given radius. Thus, arbitrary values of K_1 and K_2 may be assumed, and the values of S and D thus found may be plotted against the tangential velocity.

The S and D curves, as shown in Fig. 1, are plotted in lb. per sq. in. for one coordinate against circumferential disk velocity in ft. per sec. for the other rectangular coordinate.

For a parallel sided disk rotating at some particular speed, if S and D are known at one radius, the curves indicate what S and D will be at any other radius.

Since the equations for the S and D curves include terms for density and Poisson's ratio, the character of the curves depend on the type of material for which they were drawn. Curves drawn for steel, like those in Fig. 2, do not apply directly to materials having a different density or

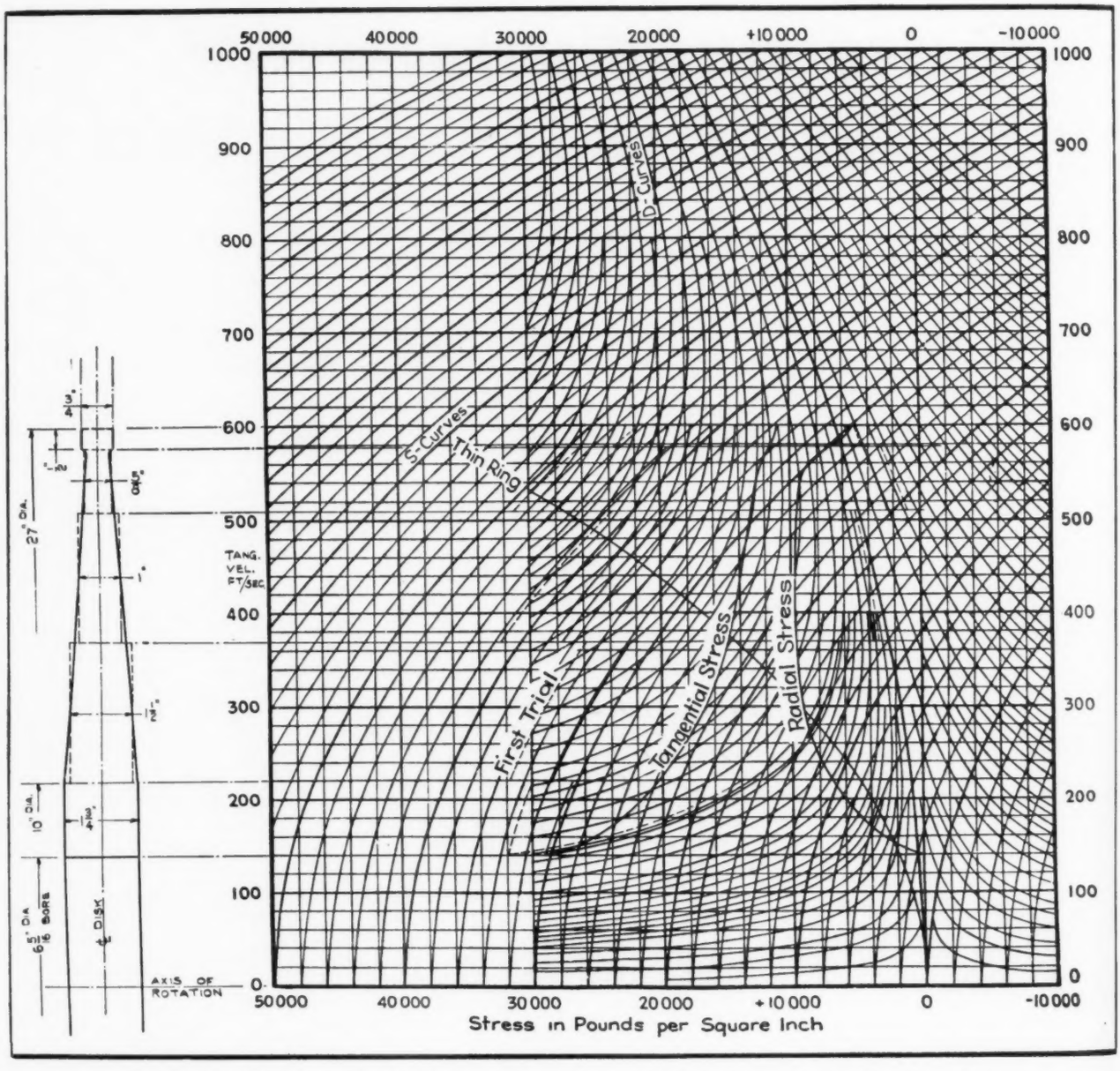


Fig. 2—H. Haerle's sum and difference curves for calculating stresses in rotating steel disks.

Poisson's ratio. Sometimes it is possible to use the curves for steel and make corrections to apply the results to another material. For aluminum, as well as for a number of other commonly used materials, Poisson's ratio may be nearly the same as for steel. Since the stress varies directly as the density, the density correction is simple to apply.

An aluminum wheel can be calculated as if it were made of steel including the rim load; the stresses are then reduced in the ratio of their densities, 0.10 to 0.28. When stresses for the steel disk are too large to fit on the *S* and *D* chart, it may be necessary to calclulate for a lower speed than the true speed. Once the stresses are determined for the aluminum wheel at the lower speed, they can be corrected for the higher speed by multiplying by the square of the speed ratio.

An additional curve that is usually shown on the Sum and Difference curve sheet is the "free ring stress" plotted against circumferential velocity. The free ring stress is the tangential stress that would occur in a thin hoop rotating at a certain speed. Such a ring would have no radial stress. Although it is only approximated in real structures, the free ring often serves to give a quick idea as to the feasibility of some design such as a flywheel or turbine drum.

Disks Without Central Holes

Aside from disks with central holes, there are two other principal types: (1) Solid disks mounted on a shaft by rabbetting or other means, and (2) disks that are integral with the shaft. In the first type, the tangential and radial stresses are assumed to be equal at the axis of rotation. Thus the *D* curve starts from zero where the tangential velocity is zero. When the *D* curve is zero, the tangential stress *T* is half of the *S* curve.

In the second type, the tangential and radial stresses are assumed to be equal at the diameter where the disk joins the shaft. Thus if the disk shown in Fig. 2 did not have a central hole, but was integral with a 6 5/16 in. dia. shaft, *D* would be zero where the tangential velocity equals 140 ft. per sec., instead of *S* and *D* being equal.

The maximum tangential stress in a solid disk of ductile material is less important than in a shrunk-on disk with its accompanying problem of loosening. Even if the tangential stress should exceed the yield point of the material, no harm will necessarily result. Some local yielding

within the disk is perfectly acceptable. In fact, peak stresses will be relieved by very small deformations resulting in a redistribution of stress throughout the disk. If local yielding becomes too extensive then unbalance, rough running and possible wrecking of the machine may occur.

The *S* and *D* method assumes elastic behavior of the disk so that after yielding begins different techniques are required to determine the stress distribution in the disk. This subject has been treated by Stodola and recently in an excellent paper by A. Stanley Thompson, "Stresses in Rotating disks at High Temperatures," *Journal of Applied Mechanics*, March, 1946. The *S* and *D* method does show the speed at which yielding will begin, which is used as a design criterion in many applications.

Disks of any shape can be closely approximated by a series of parallel sided concentric disks, after which the values of *S* and *D* for each parallel sided disk can be read from the standard curves. A relatively simple set of rules permits the engineer to shift from one disk to another. In using these curves, the stresses at any radius are assumed to be constant in the direction of the axis of rotation, that is, the disk section is reasonably symmetrical about some central plane. In an extremely unsymmetrical disk this would not be strictly true, but this condition is seldom encountered in practice.

With a wheel assembly comprising the disk proper and blades radially attached to its periphery, the first consideration is to distinguish between body stresses caused by the material in the disk itself and those caused by external loads applied to the disk.

Attached blades obviously do not constitute a continuous ring, and so must be treated as a "dead load," that is, one that cannot carry itself at any speed of rotation because it is incapable of resisting tangential stress. This is also true of the portion of the wheel rim that has been slotted to receive the blades and, in general, of the blade shrouding and any lashing wire. The disk itself should be considered as extending radially only to the bottom of the slots or grooves used for fastening the blades.

The total centrifugal force of all elements classed as dead load is calculated and applied as a radial load to the disk.

A convenient expression for use in calculating centrifugal force is:

F_c = centrifugal force, lb.
V = volume of element, cu. in.
d = diameter of center of gravity of element, in.
N = rotational speed, r.p.m.
k = a constant. k equals 4 for steel with a density of 0.281 lb. per cu. in.

$$F_c = kVd\left(\frac{N}{1{,}000}\right)^2 \tag{5}$$

At a speed of 5,100 r.p.m., the centrifugal force of the bucket fastening, buckets and shroud, F_{cf}, F_{cb} and F_{cs} respectively, on a wheel like that shown in Fig. 3 is

$$F_{cf} = 4 \times \frac{3}{4} \times \frac{11}{16} \times \pi \times 27.688^2 \times \left(\frac{5{,}100}{1{,}000}\right)^2 = 129{,}200 \text{ lb.} \tag{6}$$

$$F_{cb} = 4 \times 0.106 \times 3 \times 150 \times 31.375 \times \left(\frac{5{,}100}{1{,}000}\right)^2 = 155{,}500 \text{ lb.} \tag{7}$$

$$F_{cs} = 4 \times \frac{1}{8} \times \frac{13}{16} \times \pi \times 34.500^2 \times \left(\frac{5{,}100}{1{,}000}\right)^2 = 39{,}600 \text{ lb.} \tag{8}$$

Dead load centrifugal force
$= 129{,}200 + 155{,}500 + 39{,}600$
$= 324{,}300$ lb. (9)

The total dead load centrifugal force is then divided by the area of the outside surface of the disk proper to obtain the radial stress at the rim. Thus for the turbine wheel shown in Fig. 2, having buckets, shroud and fastenings as shown in Fig. 3, if the centrifugal force of blades, shroud and slotted part of rim is 324,300 lb., the radial stress R_r at the rim caused by the total dead load centrifugal force is

$$R_r = \frac{324{,}300}{\pi \times 27 \times 0.75} = 5{,}100 \text{ lb. per sq. in.} \tag{10}$$

Conditions at Inner Boundary. For disk with a free bore, like that shown in Fig. 2, the radial stress R_b at the bore is zero since there is no force at this point to create radial stress. Hence for a disk with a free bore, at the bore

$$S_b = D_b = T_b \tag{11}$$

Hub Stress at Bore. The tangential stress at the hub is generally the most important stress. In most designs it is the highest stress and has a vital bearing on the tightness of the disk on the shaft. Without going into an elaborate discussion of bursting strength of disks, which occurs at stresses far beyond the elastic stresses determined by the *S* and *D* curves, it can be assumed that the practical limiting speed is the one at which the wheel stretches enough to become loose on the shaft. Under this condition, the rotor could become so seriously unbalanced as to preclude satisfactory operation, even

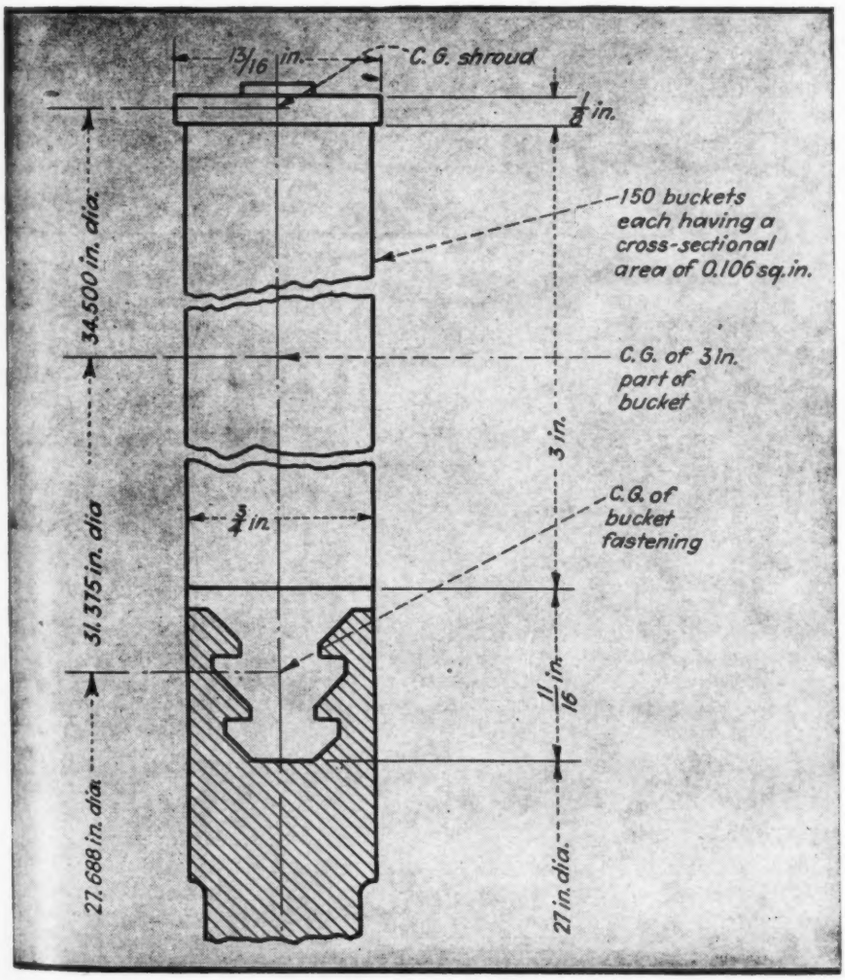


Fig. 3—Dimensions of buckets, shroud and fastening on a turbine wheel.

though the disk had not failed structurally.

The stress at which the hub becomes loose is determined by the degree of shrink or press fit and the modulus of elasticity of the material. For a shrink of 0.001 in. per in. of bore diameter the compressive stress in steel at normal temperatures would be 30,000 lb. per sq. in. When the tangential stress at the bore caused by centrifugal force exceeds 30,000 lb. per sq. in., the wheel becomes loose on the shaft.

The S and D curves give the stresses caused by centrifugal force, but not the effect of shrink stresses. This is perfectly valid, since the criterion of interest is the loosening of the disk on the shaft. When this occurs, the shrink stress is entirely relieved.

Step-by-Step Procedure in Calculating Disk Stresses.

1. Lay out the disk with radial dimensions plotted to the velocity scale of the S and D chart. Thus with the disk dimensions in inches and speed of rotation N in r.p.m., any diameter d is plotted as a velocity U, ft. per sec., equal to

$$U = \pi \times \frac{d}{12} \times \frac{N}{60} = 0.00436\, d\, N \qquad (12)$$

The width of the disk can be plotted to any convenient scale. The wheel in Fig. 2 is plotted for a speed of 5,100 r.p.m.

2. Replace the actual disk profile by a stepped profile forming a series of concentric parallel sided disks that approximate the original shape. Determine the thickness of these disks. The number of individual disks taken depends on the complexity of the original shape and the degree of accuracy required. It will rarely be necessary to take more than five or six separate disks.

3. Project the inside and outside diameters of each disk horizontally across to the S and D curve sheet.

4. Using a form similar to that of Table I enter the values

U, tangential velocity, ft. per sec., for each ring diameter.
t, axial thickness of wider individual disk at surface having tangential velocity U.
t', axial thickness of narrower individual disk at surface having tangential velocity U.

$\left(\frac{t}{t'} - 1\right)$, thickness compensation factor

5. Estimate a value of tangential stress T_b at the axis or bore of the disk as influenced by the type or method of assembly. For a first trial, T_b for the wheel shown in Fig. 2 is estimated to be 32,000 lb. per sq. in. With the estimated value of T_b compute values of S_b and D_b at the bore diameter. Enter these values in the first row of columns 5 and 6 of Table I.

6. Enter the S and D curves with the computed values at the inner diameter of the innermost ring. From this point draw lines paralleling the S and D curves until the lines intersect a horizontal line extended from the outer diameter of the first disk. Enter these values of S and D for this diameter in the second row of columns 5 and 6 of a form similar to that of Table I. Values for S and D at this diameter for the second disk are found by calculation thus:

The radial stress at the outer diameter of the first ring from Equations (1) and (2) is

$$R = \frac{S - D}{2}$$

The radial load is assumed to be transmitted unchanged across the boundary between the disks, and the radial stress to vary inversely as the disk thickness, thus the change in radial stress ΔR is

$$\Delta R = \frac{Rt}{t'} - R = R\left(\frac{t}{t'} - 1\right)$$

$$= \left(\frac{S - D}{2}\right)\left(\frac{t}{t'} - 1\right) \qquad (13)$$

Since the change in tangential stress ΔT equals the change in radial stress ΔR times Poisson's ratio, for steel with a Poisson's ratio of 0.3

$$\Delta S = \Delta R + \Delta T = \Delta R + \nu \Delta R = 1.3 \Delta R \qquad (14)$$

$$\Delta D = \Delta T - \Delta R = \nu \Delta R - \Delta R = -0.7 \Delta R \qquad (15)$$

Therefore

$$S' = S + \Delta S \qquad (16)$$

$$D' = D - \Delta D \qquad (17)$$

When the outer disk is wider than the inner disk, ΔS and ΔD will be negative, so that S' will be less than S, and D' will be greater than D. When the inner disk

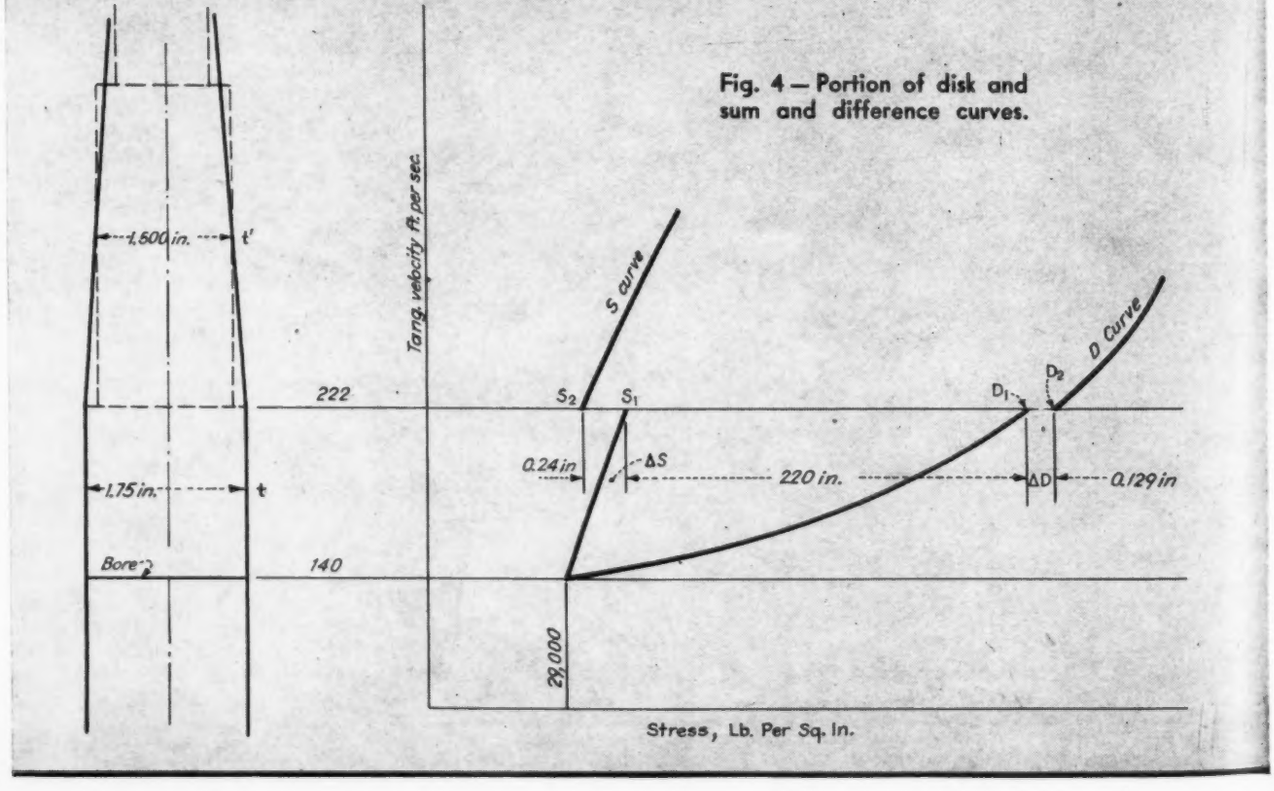


Fig. 4 — Portion of disk and sum and difference curves.

Table 1—Calculations for Stresses in Rotor Disk

For disk shown in Fig. 2, when rotating at 5,100 r.p.m. with a radial stress at rim of 5,100 lb. per sq. in.

1	2	3	4	5	6	7	8	9	10	11
U, ft. per sec.	t, in.	t', in.	$\left(\frac{t}{t'}-1\right)$	S, lb. per sq. in.	D, lb. per sq. in.	$\Delta R = \left(\frac{S-D}{2}\right)\left(\frac{t}{t'}-1\right)$	$\Delta S = 1.3\Delta R$	$\Delta D = -0.7\Delta R$	$S' = S+\Delta S$	$D' = D-\Delta D$
First trial assume $S = D = 32{,}000$ lb. per sq. in. at bore										
140	1.750	.....	.....	32,000	32,000	.....	.....	.....	.....	.....
222	1.750	1.500	0.168	30,000	14,000	1,340	1,740	−940	31,740	13,060
368	1.500	1.000	0.500	26,000	6,900	4,775	6,200	−3,340	32,200	3,560
508	1.000	0.625	0.600	24,000	5,000	5,700	7,400	−3,990	31,400	1,010
578	0.625	0.750	−0.167	26,000	3,000	−1,920	−2,500	−1,350	23,500	4,350
600	0.750	.....	.....	21,800	5,100	.....	.....	.....	.....	.....
Second trial assume $S = D = 29{,}000$ lb. per sq. in at bore										
140	1.750	.....	.....	29,000	29,000	.....	.....	.....	.....	.....
222	1.750	1.500	0.168	27,000	12,900	1,190	1,550	−830	28,550	12,070
368	1.500	1.000	0.500	22,400	6,500	3,975	5,170	−2,780	27,570	3,720
508	1.000	0.625	0.600	19,100	5,500	4,080	5,300	−2,850	24,400	2,650
578	0.625	0.750	−0.167	19,500	4,200	1,280	−1,660	+900	17,840	5,100
600	0.750	.....	.....	16,000	5,800	.....	.....	.....	.....	.....

Calculated $T_r = \frac{S+D}{2} = \frac{21{,}800+5{,}100}{2} = 13{,}450$ lb. per sq. in.

Calculated $R_r = \frac{S-D}{2} = \frac{21{,}800-5{,}100}{2} = 8{,}350$ lb. per sq. in. But $R_r = 5{,}100$

Calculated $T_r = \frac{16{,}000+5{,}800}{2} = 10{,}900$ lb. per sq. in.

Calculated $R_r = \frac{16{,}000-5{,}800}{2} = 5{,}100$ lb. per sq. in. Checks the R_r imposed on rim by buckets, shrouds, slotted disk, and rim.

is wider than the outer disk, the S and D curves get further apart, whereas if the step is from a narrower to a wider disk, the curves get closer together. This is obvious if it is remembered that the distance between the S and D curves is twice the radial stress. At the diameter where the step in thickness occurs, the thinner disk has the larger radial stress, and the thicker disk has the smaller radial stress.

Values of ΔR, ΔS, ΔD, S' and D' are next calculated and tabulated in columns 7 through 11 of Table I.

7. With the values of S' and D' thus established, follow the S and D curves to the outer diameter of the second ring, which is the inner diameter of the third ring. Then the procedure of step 6 is repeated, and the process continued for each ring until the outer rim is reached.

8. From the values of S and D at the rim, the values of the tangential stress T and the radial stress R at the rim are determined from

$$T = (S + D)/2 \quad (18)$$

$$R = (S - D)/2 \quad (19)$$

The value of R at the rim thus determined should equal the value established by Equation (10) for the radial stress R_r at the rim caused by the total dead load centrifugal force of the buckets, bucket fastenings, and shroud. If the two values are not equal, a new value for the tangential stress T at the hub must be assumed and the calculations repeated. If they still are not equal, further trials must be made until the two values coincide.

9. Once the S and D curves for the whole disk have been established, the values of T can be plotted readily. At the mean radius of each ring, mark the midpoint between the S and D curves. These points, together with T at the bore and at the rim, determine the tangential stress throughout the disk. A smooth curve should be drawn connecting all these points.

The values of R can be plotted readily by using the zero stress line as reference and stepping off half the distance between the S and D curves, again at the mean radius of each ring. These points, together with the values of R at the hub and at the rim, establish the radial stress. Again, a smooth curve should be drawn connecting these points.

The tabular method of passing from one flat sided disk to the next has been simplified by M. G. Driessen (see "A Simplified Method of Determining Stresses in Rotating Disks", *Journal of Applied Mechanics*, Paper No. APM-50-10), who also showed a means of reducing the number of trial solutions to a maximum of two.

The procedure can be explained by means of an example, again employing the disk shown in Fig. 2. In Fig. 4 a portion of this disk is shown and also a part of the S and D curves.

Proceed in the same manner as outlined for using the tabular form by assuming a value of tangential stress T_r at the bore to be equal to S and D, and for the example in question to be equal to 29,000 lb. per sq. in. Then from the bore proceed as before along the S and D curves to the end of innermost section. Call the values at this point S_1 and D_1. Since the radial stress at any section is proportional to S minus D, (actually one half of S minus D), the radial stress will be proportional to the scaled distance $\overline{S_1D_1}$. The total radial load is transmitted unchanged from the boundary of one section to the next, but the radial stress is assumed to vary inversely as the thickness; thus:

$$\overline{S_2D_2} = \overline{S_1D_1} \times (t/t')$$

Rather than find S_2D_2, simply determine ΔS and ΔD from the relations

$$\Delta\overline{S} = \overline{S_2S_1} = 0.65\ \overline{(S_1D_1)}\left(\frac{t}{t'}-1\right) \quad (20)$$

$$\Delta\overline{D} = \overline{D_2D_1} = -0.35\ \overline{(S_1D_1)}\left(\frac{t}{t'}-1\right) \quad (21)$$

Thus for the example where

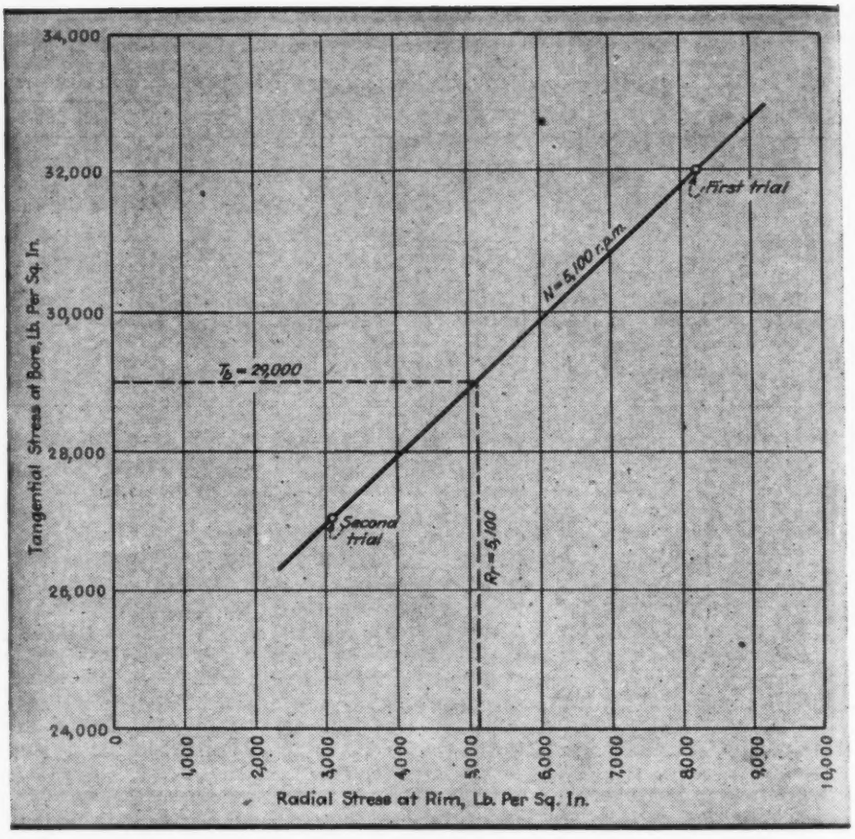
$\overline{S_1D_1} = 2.20$ in.
$t = 1.75$ in.
$t' = 1.50$ in.

$$\Delta\overline{S} = 0.65 \times 2.20 \times \left(\frac{1.75}{1.50}-1\right) = 0.24 \text{ in.}$$

$$\Delta\overline{D} = -0.35 \times 2.20 \times \left(\frac{1.75}{1.50}-1\right) = -0.129 \text{ in.}$$

By laying off $\Delta\overline{S}$ and $\Delta\overline{D}$ from the S and D curves, using the same scale as that

Fig. 5—Linear relation obtained with straight line chart constructed from two trial values of tangential stress at bore and radial stress at rim.



used in measuring $\overline{S_1D_1}$, points S_2 and D_2 are located. Then proceed along the S and D curves again to the next flat-sided disk, repeating this procedure until the rim of the disk is reached. Having determined S and D at the rim, values of T_r and R_r are calculated as previously explained.

After making two trials, the proper value of the tangential stress T_b at the bore can be determined that gives the known radial stress at the rim. M. G. Driessen has shown that a linear relationship exists between the tangential stress T_b at the bore and the radial stress R_r at the rim. Thus, by making two trials a straight line graph can be established using the two trial values of T_b and R_r. From this graph, knowing the actual value of R_r, the proper value of T_b can be found.

For example, in the disk shown in Fig. 2 if on the second trial it had been assumed that T_b was 27,000 lb. per sq. in. (instead of the correct value of 29,000 lb. per sq. in.), the trial value of R_r would have been found to be 3,100 lb. per sq. in. On a straight line chart, Fig. 5, constructed by use of the two trial values of T_b and R_r, for a value of R_r equal to 5,100 lb. per sq. in., the correct value of T_b is equal to 29,000 lb. per sq. in. Thus from two sets of values determined by trial and error, a simple linear relation is obtained.

The use of elastic stresses in design should be clearly understood. Where local deformation in a disk must be held below some specific maximum amount, as at the bore, or in a centrifugal impeller at a sealing ring, it is important to know and to control the elastic stresses. It is also important to know and to control the elastic stresses when fatigue is a factor or in very brittle materials. Where these conditions do not apply, the average tangential stress in the disk is the important design criterion, and elastic stresses have only academic interest. The average stress can be found from S and D curves, but it can also be found more quickly by a direct calculation.

Case Histories In Patent Law

H. A. TOULMIN, JR.

[Editor's Note: These explanations of the law in specific patent cases are not intended to supplant the services of a patent attorney, whose accurate advice is necessary to handle individual patent problems. The purpose of these explanations is to give a better general understanding of patent law.]

NO. 5

QUESTION: Under what circumstances will the application of an old process to a new and analogous use be lack of invention?

ANSWER: In this case a process patent was before the Supreme Court of the United States involving the addition of an inhibiting agent to hydrochloric acid solution to prevent corrosion, the use of a dilute rather than a concentrated hydrochloric acid solution, and the use of an ordinary pump tube instead of a specially protected supply pipe to introduce acid into the well.

Inhibiting agents to prevent the corrosive effect of acid on the metal were old. It was well-known that hydrochloric acid would dissolve limestone and increase the production of oil wells. It was well-known that hydrochloric acid would corrode metal and that arsenic compounds and other chemicals could be added to hydrochloric acid to stop its corrosive effect. It was well-known that inhibited hydrochloric acid could be effectively utilized to remove scale from metal well equipment without corroding the metal.

The Supreme Court found that the process merely taught the obvious fact that the elements that constituted steps in the law of nature of hydrochloric acid could be utilized to inhibit and prevent corrosion while being used to dissolve limestone rock pursuant to the old method of acidizing wells. Such a process lacked the very essence of invention.

The mere fact that no one had apparently thought to use an inhibitor while acidizing an oil well to increase production could not lift the process under consideration to the status of an invention. It is necessary to do more than to put a number of elements together which have no cooperative effect upon one another—the mere substitution of more effective equivalents that were well-known in the arts. No new function was performed by the new combination and no new results were secured. Dow Chemical Co. v. Halliburton Oil Well Cementing Co., 324 U. S. 320.

NEW BOOKS

The Technique of Executive Control

ERWIN HASKELL SCHELL. *Sixth Edition*, 270 pages. 5¼ x 7½ in., blue clothboard covers. *Published by the McGraw-Hill Book Co., 330 W. 42nd St., New York 18, N. Y. Price* $2.25.

This text is of direct and practical use to the executive in his relationship with subordinates, associates, and superiors. The fundamental principles and basic rules of action generally applicable in these relationships are discussed in detail. A general bibliography for additional reading is included together with thought-provoking questions interspersed throughout the text. The sixth edition expands the original presentation to include new applications of executive technique in the fields of executive attitude, morale, and public relations.

How to Supervise People

ALFRED M. COOPER. *Second Edition*. 162 pages. 5 x 7½ in., blue clothboard covers. *Published by the McGraw-Hill Book Co., Inc., 330 W. 42nd St., New York 18, N. Y., Price* $2.

A manual of the principles of industrial supervision. This edition has been revised and brought up to date to be of particular value to those directing the activities of the industrial employee. It explains how supervisory employees can analyze their jobs and apply correct methods of handling problems encountered in their work. Giving special attention to postwar supervisory difficulties, this edition adds a new chapter on the industrial supervisor's responsibility for developing good public relations for his company. It includes revisions and additions to the questions for group discussion included in the text.

The Physically Handicapped Worker in Industry

GILBERT BRIGHOUSE. 54 pages. 6 x 9 in., gray paper covers. *Published by the Industrial Relations Section, California Institute of Technology, Pasadena 4, Calif. Price* $2.

A report of an investigation into the effectiveness of the physically handicapped employees of the Lockheed Aircraft Corporation in Burbank, Calif. The aim of this study was to determine as objectively as possible the characteristics of these employees, their job performance, absences, accidents, and morale in comparison with a group of physically normal employees selected at random. For practical reasons, several limitations were imposed on the scope of the study. All data were gathered between October 1944 and January 1945 and the investigation is concerned only with those employees who were working during that period or who had recently left the company. Employees with obvious mental handicaps were excluded as were those with temporary disabilities. Results of this investigation indicate that the physically handicapped worker is more likely to be involved in an accident than the able-bodied employee. These accidents are more serious in terms of time lost and compensation paid. Flexibility of employees is somewhat lost in the case of the physically handicapped, as more time must be spent in placing them correctly, and their quality and quantity of output is slightly lower than that of normal employees. They do, however, show greater stability on the job, less readiness to quit, less unexcused absenteeism, and a great willingness to stay put on certain jobs without possibilities of promotion to supervisory status.

Experimental Stress Analysis

Proceedings of the Society for Experimental Stress Analysis, Vol. III, No. 2 166 pages, 8½ x 11 in., blue clothboard covers. Published by Addison-Wesley Press, Inc., Kendall Square Building, Cambridge 42, Mass. Price $5.

Collection of 12 papers. Titles of papers with respective authors are:

Improvements in Rosette Computer. J. H. Meier.

New Methods of Analysis of Machining Processes. M. E. Merchant and N. Zlatin.

Stress Distribution in Spur Gear Teeth. F. H. Boor and E. O. Stitz.

A Study of the Mechanical Behavior of the Skull and Its Contents when Subjected to Injuring Blows. H. R. Lissner and E. S. Gurdjian.

Use of Electric Resistivity Strain Gages over Long Periods of Time. G. P. Tschebotarioff.

A Method of Effecting SR-4 Strain Gage Operation under Water. E. L. Kimble.

Bicycle Spokes—Their Use in Testing Aircraft Structures. C. R. Smith.

A review of Some Mechanical Failures of Steel Plant Machine Equipment. S. W. Poole and R. J. Johnson.

Improved Techniques and Devices for Stress Analysis with Resistant Wire Gages. W. V. Bassett, H. Cromwell and W. E. Wooster.

Allowable Working Stresses. G. C. Noll and C. Lipson.

Photogrid Strain Analysis of Formed Parts. George Gerard.

Elementary Applied Aerodynamics

PAUL E. HEMKE. 231 pages, 6¼ x 9¼ in. Red clothboard covers. *Published by Prentice-Hall, Inc., 70 Fifth Ave., New York 11, N. Y. Price* $4.35.

A text meeting the requirements of a first course in technical applied aero-dynamics. The material is claimed to have been successfully presented for a number of years in the author's classes in aerodynamics at Rensselaer Polytechnic Institute. Certain topics covered in this text are generally found in more advanced courses. Among these is a more rigorous treatment of the stream function of an ideal fluid flow. A short chapter on compressibility effects and an introductory chapter on rotors and helicopters are included. Illustrative examples are used throughout and problems are included at the end of each chapter.

The Heating of Steel

M. H. MAWHINNEY. 265 pages, 6¼ x 9¼ in., maroon clothboard covers. *Published by the Reinhold Publishing Corp. 330 W. 42nd St., New York, 18 N. Y. Price* $5.

A practical discussion of the methods and equipment necessary to obtain the best results when heating steel. The chemical effects of steel under heat as well as the fuels and burner equipment used are discussed. Temperature distribution, furnace control, heat transfer, and fuel economy are included, in addition to material on quenching, alloys and refractories. Considerable space is devoted to the design, construction, and operation of various types of steel mill furnaces.

Engineering Mechanics

SEIBERT FAIRMAN AND CHESTER S. CUTSHALL. *Second Edition*, 268 pages, 6 x 9¼ in., gray clothboard covers. *Published by John Wiley and Sons, Inc., 440 Fourth Ave., New York 16, N. Y. Price* $3.

In the preparation of this edition, no radical departures from the original text have been made. A chapter on product of inertia and moments of inertia about principal axes has been added as well as new descriptions and problems. The test is presented in the same pattern as the original. In recognition of the importance of dimensional analysis and its applications, a brief introduction to this subject has been made at the beginning of that part of the book dealing with kinetics, and the dimensional formula is given for each new quantity taken up thereafter.

X-Rays in Practice

WAYNE T. SPROULL. 615 pages 5¼ x 8½ in., 24 chapters, blue clothboard covers. *Published by McGraw-Hill Book Co., 330 W. 42nd St., New York 18, N. Y., Price* $6.

A fundamental text on the application of X-rays particularly in fields other than medical, is provided in this book. The chapter arrangement is topical for easy reference use, yet it carries the student logically from history and the fundamentals of X-ray tubes through the continuous X-ray spectrum, characteristic spectro, generating equipment to the rudiments of radioactivity, X-ray detention, measurement, and protection. Medical use of X-rays is only briefly described because the techniques are largely qualitative rather than quantitative. Industrial radiography is more fully covered.

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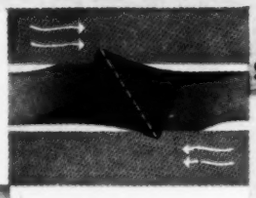

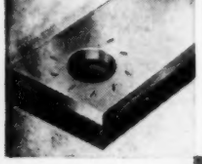

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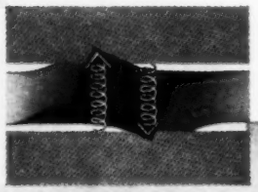

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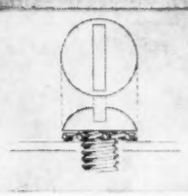

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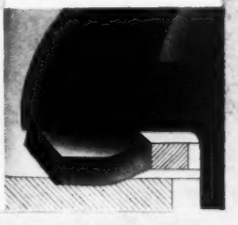

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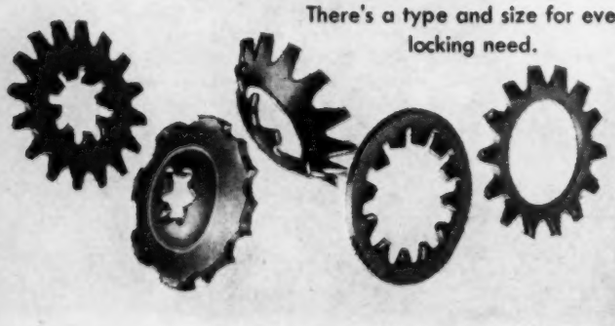

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Chart for Efficiencies of Worm Drives

JOHNSON H. K. KAO
Timken Roller Bearing Company

THE ACCOMPANYING CHART, page 157, is used to determine the approximate efficiencies of worm drives and gives a graphical solution to the formulas:

$$\text{Efficiency} = \frac{\tan \text{helix angle}}{\tan (\text{helix angle 1 deg. 4 min.})} \quad (1)$$

$$\text{Tan helix angle} = \frac{D_G}{D_W} \times \frac{1}{R} \quad (2)$$

where D_G = pitch diameter of worm gear, in.
D_W = pitch diameter of worm, in.
R = worm drive ratio

The following examples are solved on the chart (see dashed lines).

Example: What is the approximate efficiency of a worm drive, when

Worm drive ratio = 12:1
Worm pitch diameter, $D_W = 1.75$ in.
Worm wheel diameter, $D_G = 6.25$ in.

Solution: $\frac{D_G}{D_W} = \frac{6.25}{1.75} = 3.57$

At point 3.57 on the abscissa, erect a perpendicular to meet diagonal R equals 12 at point A. Follow horizontally from A to meet tangent curve at point B, then follow vertically to meet efficiency curve at point C. Read 93.4 percent from efficiency scale.

Example: What is the approximate efficiency of a worm drive when the helix angle equals 16 deg. 35 min.?

Solution: At point 16 deg. 35 min. on the abscissa, erect a perpendicular to meet the efficiency curve at point C. Read 93.4 percent from efficiency scale.

It will be noted that the helix angle of any worm can be determined from the given pitch diameter on the chart, and the ratio of drive obtained for the calculation of power transmitted. The chart can also be used inversely for the design of worms and wheels of maximum economical efficiency with given ratio of drive.

DERIVATION OF FORMULA (1)

In helical gearing,

f = coefficient of friction
a = arctan f
b and c = respective spiral angles of the pair
$b + c$ = shaft angle

hence, $\text{Efficiency} = \frac{\cos b \times \cos (c + a)}{\cos c \times \cos (b - a)}$

In worm gearing, shafts are normal to each other.

$b + c = 90$ deg. and $\cos b = \sin c$
$\cos (b - a) = \cos b \cos a + \sin b \sin a$
or, $\cos (b - a) = \sin c \cos a + \cos c \sin a = \sin (c' + a)$

hence, $\text{Efficiency} = \frac{\sin c}{\cos c} \times \frac{\cos (c + a)}{\sin (c + a)}$

$$= \frac{\tan c}{\tan (c + a)}$$

Values of f vary from 0.002 to 0.06 for different conditions of lubrication and speed. For average industrial applications, f equals 0.0185, which is equivalent to a friction angle of 1 deg. 4 min.

DERIVATION OF FORMULA (2)

Since $\tan \text{helix angle} = \frac{\text{lead}}{\pi D_W}$

and $R = \frac{\pi \times D_G}{\text{lead}}$

Equate the two leads and cancel π

$\therefore \tan \text{helix angle} = \frac{D_G}{D_W} \times \frac{1}{R}$

Hints for Product Designers and Engineers

Today, when every designer and engineer is confronted with the task of creating new peacetime products, it is of prime importance that creative effort be unhampered by detail, and that the "tools" be right.

A few moments spent now in considering the recommendations that follow will result in improved, trouble-free drawings in the busy months ahead.

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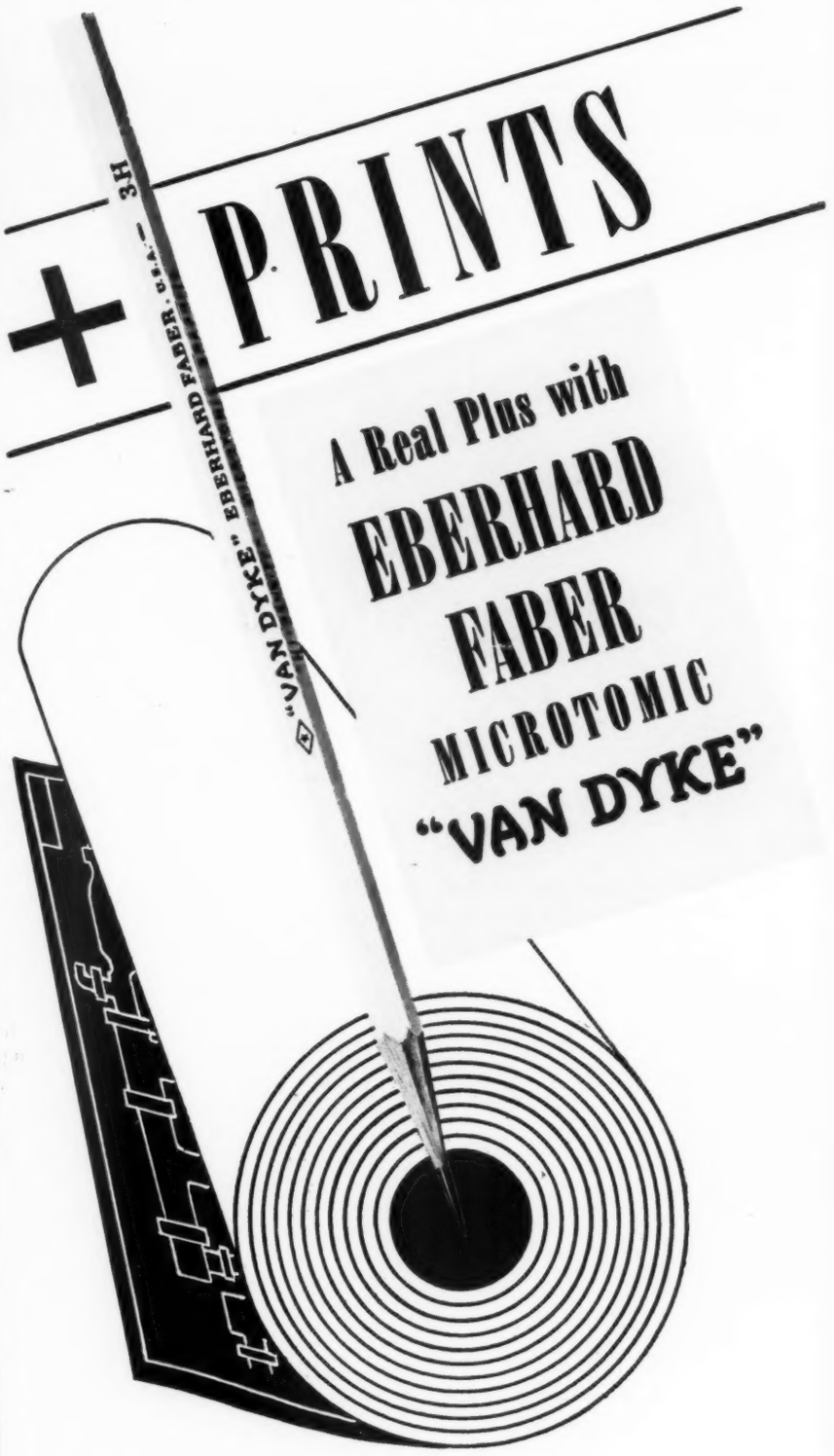

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Chart for Efficiency of Worm Drives (continued)

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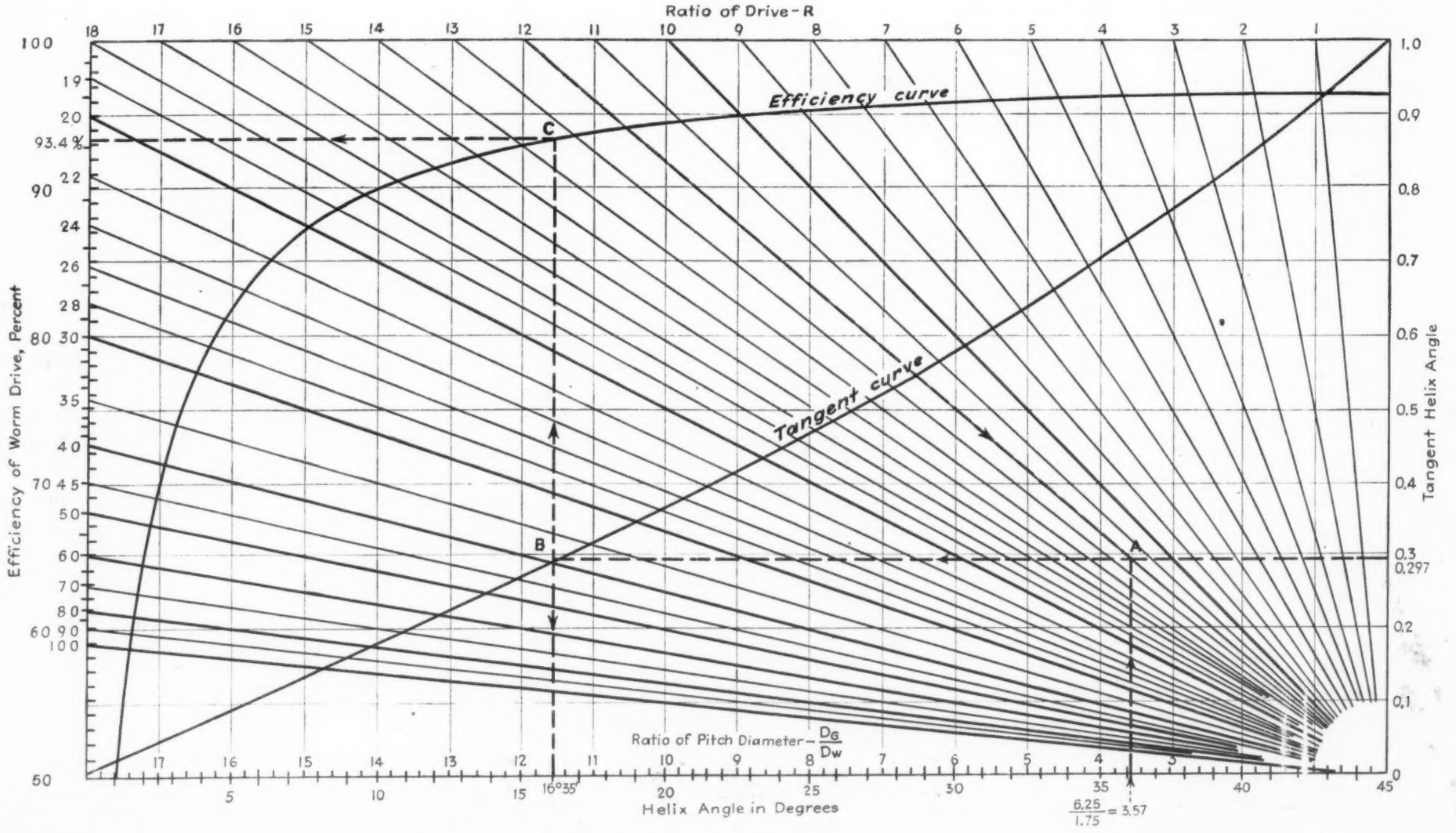

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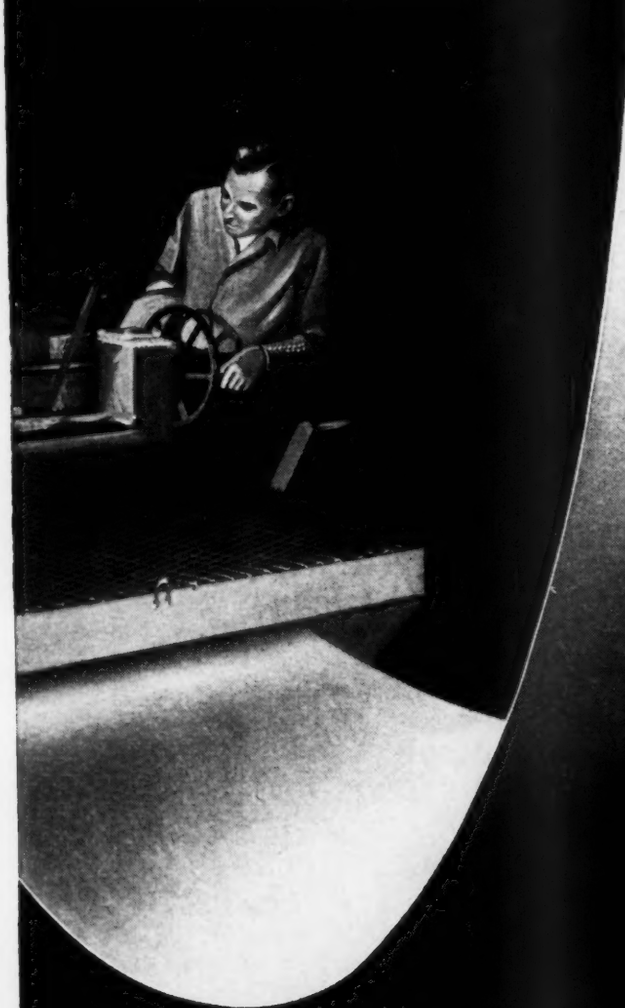

ALUMINIZED ARMCO STEEL

THE AMERICAN ROLLING MILL COMPANY

NEW MATERIALS AND PARTS

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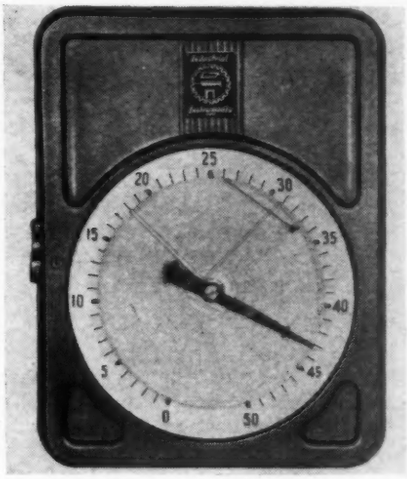
Electrolytic Conductivity Recorder

Industrial Instruments, Inc., 17 Pollock Ave., Jersey City 5, N. J.

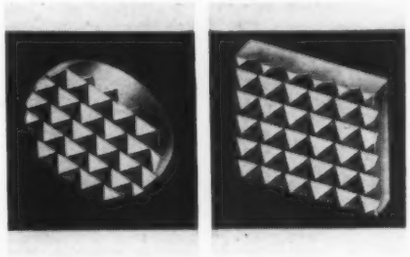
The Type RP unit records the electrolytic conductivity of solutions, condensates or water by the solubridge method. The Solubridge Recorder has an a.c. Wheatstone bridge circuit operating on the null balance principle. The accuracy is independent of line-voltage fluctuations and vacuum-tube characteristics. The instrument can be furnished for 1, 2, 3 or 4 continuous-line records on the same 12-in. dia. circular chart. A separate amplifier and motor-driven slide wire are used for each recording pen, and no switching devices are necessary. All pens of a multiple-pen recorder can move at any time or simultaneously. The amplifier is a standard and interchangeable unit. As optional equipment, the recorder can include a 29 in. indicating scale and bold indicator pointer, illuminated by an optional built-in fluorescent lamp. The indicator pointer has a magnetic lock so that operation of the chart release lever lifts both indicator pointer and recording pens, leaving the chart free for exchange. The conductivity recorder can be provided with a micro-switch mounting actuated by the same slide-wire unit that drives the recording pen. This control switch is operated by an adjustable cam set to trip at any desired position of the recording pen or indicating pointer. Four switches, individually adjusted, can be provided in the single pen recorder. Air-operated proportioning controllers can be supplied as optional equipment.



Gripping Inserts

Kennametal Inc., Latrobe, Pa.

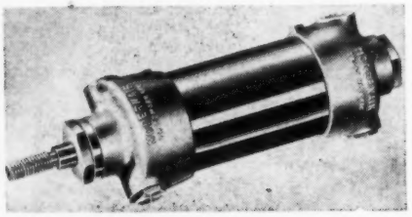
Known as "Kengrips", these gripping inserts are used in various holding, clamping, and feeding devices. "Kengrips" are made of solid Kennametal, in the forms of disks and squares, with diamond serrated surfaces. Four sizes are available; in either form, as follows: $\frac{1}{2}$ x $\frac{1}{8}$ in. thick; $\frac{5}{8}$ x $\frac{1}{8}$ in. thick; $\frac{3}{4}$ x $\frac{5}{32}$ in thick, and 1 x $\frac{5}{32}$ in. thick.



Cushion-Type Cylinders

Modernair Corp., 4222 Hollis Street, Oakland 8, Calif.

The cylinder caps are heat-treated aluminum alloy sand castings and rotate 360 deg. on the cylinder without disturbing the seal. Aircraft-type "O" ring packings are used throughout the cylinder at all



points of wear or possible leakage. No metal-to-metal contact exists at any point of wear. The cylinder is corrosion resistant, and can be actuated by air, water or oil. The cylinder caps are designed to take one of several available brackets or any combination of brackets. These mounting brackets are: End plate, foot bracket and swivel mounting. All brackets are interchangeable, and the basic cylinder can be re-used in different applications by a change in mounting method. Modernair cylinders are available in 2, 3, 4, 5 and 6 in. diameters.

Percentage Timer

The R. W. Cramer Co., Centerbrook, Conn.

Known as the Type PE, this unit is designed for use on alternating current to control automatically the percentage of time at which any electrical circuit can be closed or opened out of a definite

length of time. Calibrations and ranges are adaptable to specific process requirements. These timers will repeat within 0.25 percent of any given setting and their specially designed motors and operating switch mechanisms are rated for continuous service. They are completely inclosed in dust-proof plastic housings.

Radiation Meter

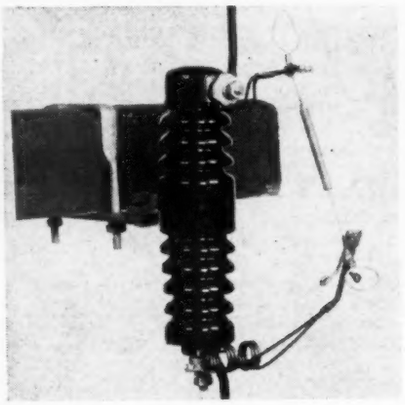
General Electric Co., Schenectady 5, N. Y.

Intended primarily for measurement of solar radiation, this instrument consists of two major components: A radiation receiver mounted in a mental inclosure on the top of the case, and a D'Arsonval type indicating instrument connected to the receiver output leads. The radiation receiver is a thermocouple of the total radiation type. It is sensitive to radiation of wave lengths from 0.3 to 3.5 microns. The indicating instrument is a permanent-magnet, moving-coil millivoltmeter. The scale length is 1.5 in. and the range of the meter is 0 to 2 gram-calories per sq. cm. per min. Accuracy is ±5 percent of full-scale value. A metal instruction plate, giving operating instructions and conversion factors for other units of measure, is fastened on the back of the case. Net weight of the meter is 6 ounces.

Fuse Cutout

General Electric Co., Schenectady 5, N. Y.

An improved design of the flip-open fuse cutout in 50-ampere, 7,500 and 15,000-volt ratings. These units have a porcelain insulator with the mounting support and



the upper and lower contact supports cemented into the porcelain. The live parts are confined to the side of the insulator opposite the mounting bracket. The cutout can be mounted vertically, angle-tilted or swiveled to any desired position.

Steam Trap

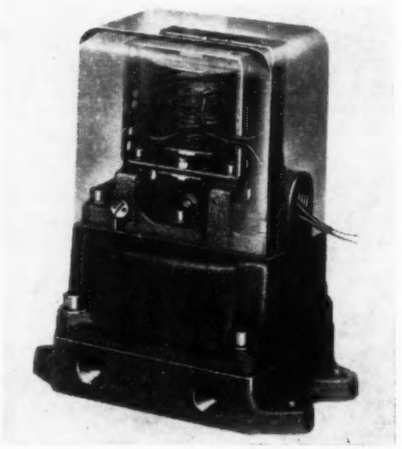
Sarco Company Inc., 475 Fifth Ave., New York 17, N. Y.

A thermostitic trap measuring 2⅜ in. high, 1 in. center to face of inlet, and weighing 13 oz. Connections are ¼ in. or ⅜ in. I. P. S. Known as Sarco No. 10, it is equipped with helically corrugated bronze bellows, brass body and renewable valve head and seat of stainless steel. Maximum operating pressure is 100 lb. per sq. in. The trap can also be used as a high pressure thermostatic air vent.

Four-Way Valve

Numatics, Milford, Mich.

A series of four-way solenoid-controlled 4-port and 5-port valves. These units can be mounted in any position with either



side or bottom pipe connections used. Two poppet elements, without springs, alternately open and close to pressure and exhaust by "Fluid Lever" air from a solenoid-operated central distributor. These poppet elements control the respective ends of a double-acting air cylinder, with either single or dual pressure. Seat ports and air passages are larger than the inside of standard pipe. Working pressures from 0 to 150 lb. per sq. in. are handled and a low amperage solenoid with 5/64 in. travel is used. The valve is available with 4-port base for single-pressure and with 5-port base for dual-pressure service. The valve body and upper structure will fit either base. Known as SR4 series, these valves are available in the following pipe sizes: ¼, ⅜, ½, and ¾ inches

Spring Locking Fastener

Lion Fastener, Inc., Honeoye Falls, N. Y.

A locking device consisting of two parts: Stud and cantilever spring assembly. The stud is of one-piece construction with shoulders forged on opposite sides to form the locking member. Studs are available with round, flush, or wing heads. The spring assembly, comprising a cantilever-type spring and base, is a pre-assembled unit, ready to install. The base forms a supporting member for the spring under extreme loads. The floating leaf spring is held by retained lugs at each end of the

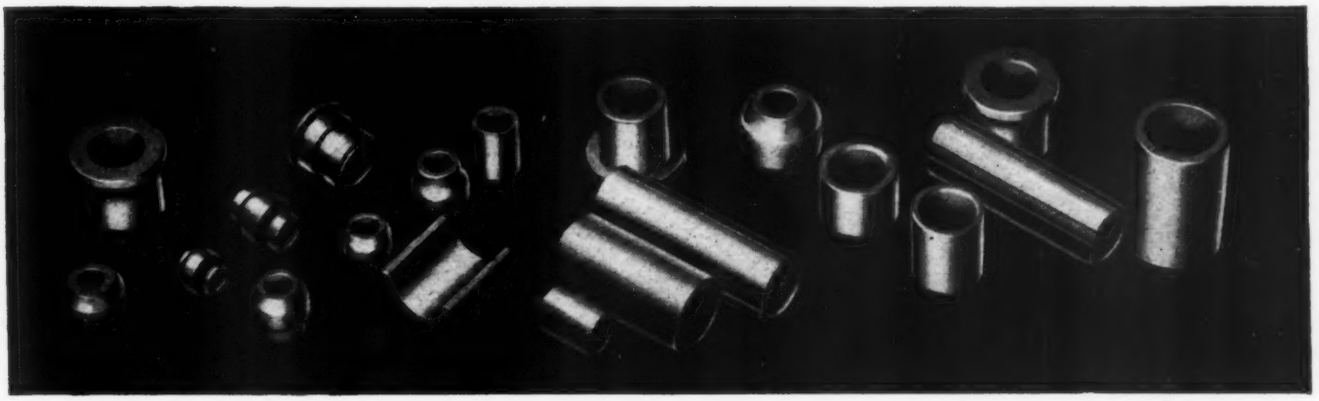
The RIGHT *amount of* OIL

in the RIGHT *place*

at the RIGHT *time*

Oil is the lifeblood of a bearing. Proper lubrication forms a protecting film between the shaft and the bearing . . . prevents metal-to-metal contact . . . cuts friction to a minimum . . . prolongs the life of the entire motive unit.

One reason why Johnson LEDALOYL Bearings operate so quiet . . . so smooth . . . and for so long a time is the self-lubricating feature. Myriads of tiny, evenly spaced pores . . . as many as 13,000 to the square inch . . . serve as miniature oil wells. When the bearing is in use these pores meter the oil to the bearing area. When the bearing is at rest the oil is reabsorbed. Thus you always have the *right* amount of oil in the *right* place, at the *right* time.



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The dependable self-lubricating action of Johnson LEDALOYL provides many worthwhile savings to the designer. Expensive lubrication aids are eliminated and, in many cases, the bearings can be sealed in place. They are ideal for all types of household equipment. Write for complete, technical information.

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BEARING BRONZE

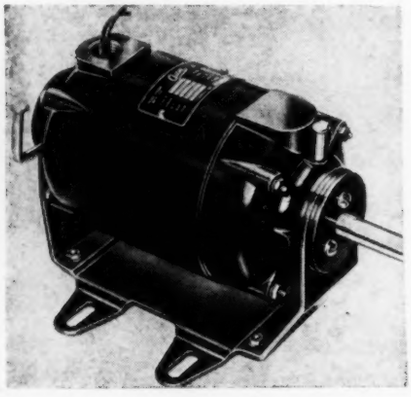
JOHNSON BRONZE
SLEEVE BEARING HEADQUARTERS
BRANCHES IN 18 INDUSTRIAL CENTERS

base, and is formed with a double set of grooves and cams that interlock with the notched shoulders of the stud. A quarter turn of the head locks the stud in the spring assembly.

Electric Motor

John Oster Mfg. Co., Racine, Wis.

The type KS-837 unit delivers to 1/10 h.p. at 1,725 r.p.m. It is a split phase induction, continuous duty model, operating on 115 volts, a.c. The motor is totally inclosed. Tubular steel stator housings are rabbetted to fit die cast end bells and the bronze sleeve bearings are grooved for oil distribution. Oversize felt oil retainers feed oil to



shaft through two spring tension wicks at each bearing. Neoprene motor mountings absorb vibration and are not affected by oil or grease. Flange mounting is also available. Length over bearing 8-3/8 in., housing diameter 4-27/64 in., weight 15 lb. Finished in black enamel.

Glass Fiber Retainer Mats

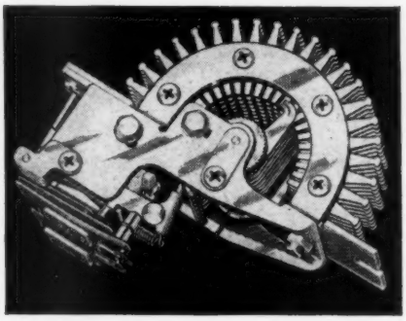
Johns-Manville, 22 East 40th Street, New York 16, N. Y.

Thin diaphragms or filter screens for inclosing the positive plates of acid type storage batteries. These glass retainers hold migrant particles of lead in the facial interstices of the glass fibers. Thicknesses are available from 5 to 100 mils in rolls 50 in. wide by 100 to 150 ft. long. Double mats are furnished with a center crease.

Spring-Driven Stepping Switch

The C. P. Clare Co., 4719 West Sunnyside Ave., Chicago 30, Ill.

Control operations, such as selection, sequence, counting, and totaling are performed by this unit. The switch consists essentially of one or more wiper springs, fixed on a shaft rotated by a pawl and ratchet mechanism. This mechanism is actuated by an electromagnet that responds to momentary current impulses. When the magnet is de-energized, the wiper assembly is driven for-



ward by a spring. Framework and armature are finished in cadmium. Bank contacts and wipers are phosphor bronze. Maximum number of contacts for a 20-point switch is 8; for a 40-point switch, 4. Maximum operating speed with remote control is 30 steps per second. Average operating speed with self-cycling operation is 60 steps per second.

Industrial Wheel

The B. F. Goodrich Company, Akron, Ohio

A wheel constructed with a metal bearing sleeve molded integrally in a hard rubber core in which ball bearings for a choice of axle diameters are mounted. Carrying capacity ratings for these all-rubber wheels are claimed to be equal to those for vulcanized-on tires and metal wheels of the same size. They are not recommended where rough usage is encountered.

Tension Brake

Linderman Devices, Inc., Newburgh, N. Y.

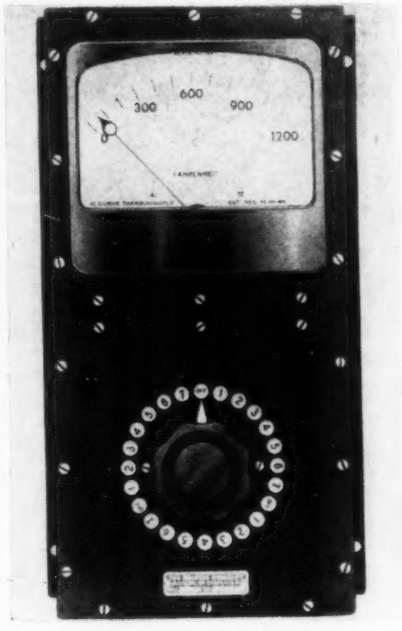
Designed to apply controllable continuous tension in industrial machinery. In operation, the desired amount of tension is obtained by setting the air pressure regulating valve in the lead from the incoming air line. Slight increases or decreases in tension are obtained by changing the gage setting, since the amount of braking effort is directly proportional

to the air pressure applied. The shoes are self-equalizing, full-floating, with no localized high pressure areas on lining or drum. Drums are finned for maximum heat-dissipation under continuous application. Fan cooling can also be provided, by inclosing the brake assembly within an annular sheet metal housing. Application of shoe pressure to the drum is through multi-stage steel diaphragms. The brake shoes have a wedge mechanism to adjust for lining wear. The adjustment screws and the shaft bearings are the only parts requiring lubrication.

Pyrometer

The Bristol Co., Waterbury 91, Conn.

The model 6-426 commercial diesel engine pyrometer has a millivoltmeter type of movement damped to resist vibration and give quick response to temperature



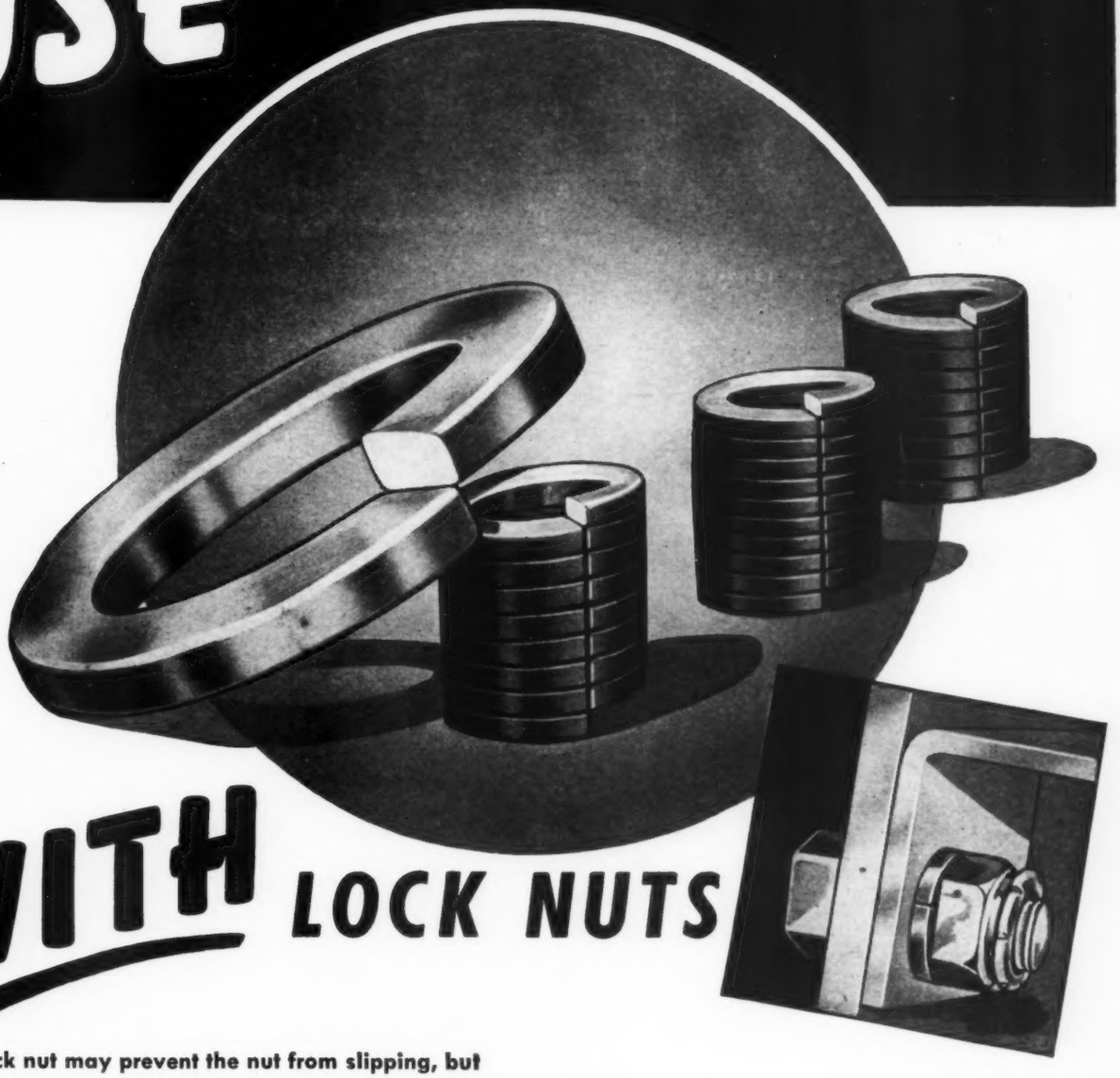
changes. A rotary switch is provided, having 24 points and a universal dial that enables it to be adapted for various combinations of engines and numbers of cylinders. Thermocouples for installation in all types of engines can be furnished.

Autotransformers

General Radio Co., 275 Massachusetts Ave., Cambridge 39, Mass.

Designated as Variac Type V-5, this unit is intended for the manual control of a.c. voltage. Proper brush pressure is assured by a coil spring, and brush extension is limited to prevent short-circuit damages through contact of the brush holder with the winding. Finish is a black matte baking lacquer. Rated output current for 115-volt models is 5 amp.; maximum current is 7.5 amp. Voltage output can be

USE SPRING LOCK WASHERS



WITH LOCK NUTS

The lock nut may prevent the nut from slipping, but it can't compensate for the wear or corrosion of the assembly parts or the stretching of the bolt. That's why you need a spring lock washer. Lock nuts with spring lock washers make an ideal combination—safeguard against loss of nut and provide protection against looseness with positive tension.

Diamond G Spring Lock Washers—scientifically designed with CONTROLLED TENSION—assure unfailing spring tension . . . plus a thrust washer bearing. They permit full tightening of bolts and screws and safeguard against excessive vibration, shock and wear.

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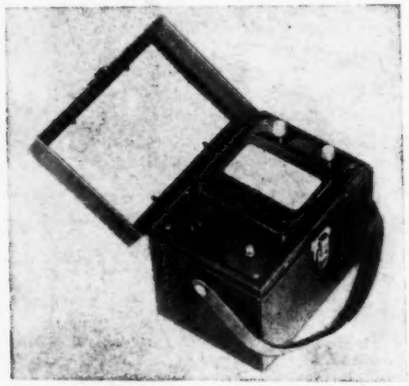
varied from 0 to 17 percent above line voltage. Volt-ampere rating is 0.862 kva. 230-volt models will handle 0.575 kva, with current ratings of 2 amp., rated, and 2.5 amp. maximum.

Insulation Resistance Meter

Associated Research, Inc., 231 So. Green St., Chicago, 7, Ill.

A portable, self-contained insulation resistance meter, testing to 50,000 megohms. The unit, Model 261 Vibrotest has a hand-stepped scale, calibrated against known standards of resistance. It has a high voltage regulator in the measuring



circuit, and is equipped with condenser charging circuit. The meter is tested for temperature variations, from −40 to +140 deg. F. It is guarded against stray leakage, and for operation in extreme humidity. Model 261 is housed in weatherproof metal case with carrying strap. It measures 8¾ x 6 x 8 in. and weighs 17½ pounds.

Pressure Gage

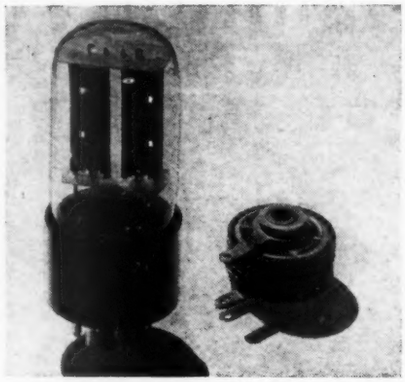
American Machine and Metals, Inc., Sellersville, Pa.

A heavy duty pressure gage with internal parts made of non-corrosive metals. The working elements are designed to withstand repeated pulsation, vibration and high overpressures. Supergages are made for pressures ranging from 30 to 10,000 lb. per sq. in. and are available 4½, 6, and 8½ in. dial sizes.

Selenium Rectifier

General Electric Co., 1285 Boston Ave., Bridgeport 2, Conn.

Consisting of five selenium disks made on aluminum base plates and connected in series, the rectifier measures 1 in. in diameter and ¾ in. long. No socket or filament circuit is necessary. It is designed to withstand the inverse peak voltages obtained when rectifying (half-wave) 110-120 volts RMS and feeding into a capacitor as required in the various radio circuits when half-wave rectification of the

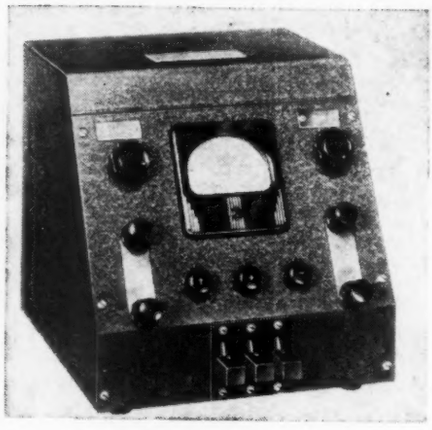


a.c. line voltage is used. Continuous currents to 100 milliamperes can be obtained with a temperature rise below the safe limits for selenium rectifiers. High momentary peak currents are not detrimental to the unit.

Limits Bridge

Associated Research, Inc., 231 S. Green St., Chicago 7, Ill.

Low Range Limits Bridge, Model 81-A, is completely self-contained. It uses three No. 6 dry cells. This internal 4½ volt battery suffices for intermittent duty for a wide



range of resistances. Provision is made with binding posts for an external battery connection where higher sensitivity is required. The unit does not have to be leveled or installed on the test bench. Resistance from a fraction of an ohm to 20,000 ohms can be checked in tolerances of ±1 to ±20 percent. Plus and minus tolerances are separately adjustable.

Variable Condensers

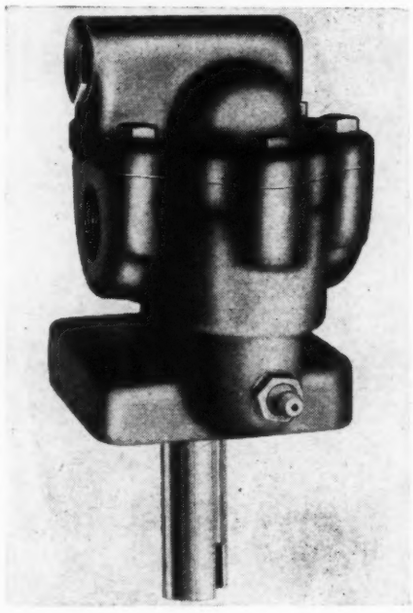
Airadio, Inc., Stamford, Conn.

Two and three gang condensers of the RMA Class B type. An oscillator section providing a maximum of 162 mmf. for tracking with a 455 kc. I. F. in the broadcast band, and an antenna section providing a maximum of 452 mmf. are available. The units have three holes at the bottom, three at the front and three at the rear, tapped 6/32. An additional four holes at the front and four at the rear, tapped 8/32, permit mounting at any of three positions. These condensers can be supplied to meet specifications requiring any of the three standard torques, 1 to 4 in. oz. 2 to 6 in. oz. or 3 to 8 in. oz. A maximum of four trimmers may be had for the two-gang unit.

Gear Pump

Gray-Mills Corp., 1948 Ridge Ave., Evanston, Ill.

An all-bronze unit for pumping water, oils, solvents, salt solutions, chemicals and other liquids of corrosive or non-corrosive nature. Known as the "GB", this pump has a



maximum pressure of 300 lb. per sq. in. at 1,750 r.p.m. Volume is 6 gal. per min. Discharge connection is 1/2 in. The pump can be operated by pulley or direct drive and can be mounted internally or externally in various positions. It is available as a pump only; a pump and motor; or as a portable pumping unit with motor and tank.

Recorder-Controller

C. J. Tagliabue Div., Portable Products Corp., 602 Park Ave., Brooklyn 5, N. Y.

An automatic recorder-controller for temperature and pressure. An adjustable cam starts the timing automatically when the temperature reaches the processing point and is terminated at the desired moment. All valves, whether steam, air, water or overflow, are opened and closed as the process requires without manual attention. A red light glows on the controller during the entire steam-heating phase of the process. When heating has been com

Specifications alone

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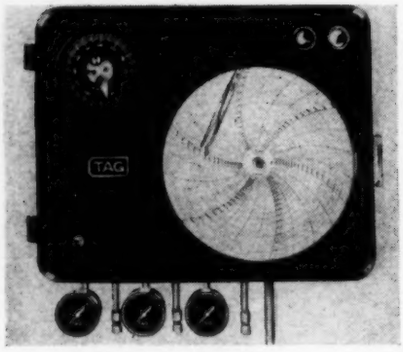
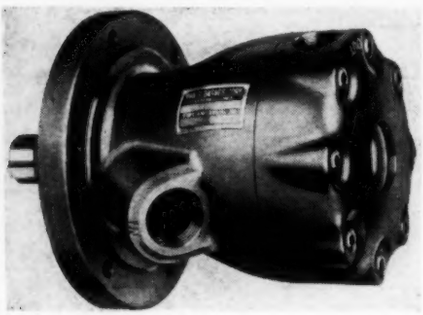


pleted and steam has been shut off, the red light is extinguished and a white light appears. After sufficient time for cooling has elapsed, the white light goes out, indicating visibly to the operator that this cycle has been completed.

Hydraulic Pump

John S. Barnes Corp., 301 So. Water St., Rockford, Ill.

Known as the Equalized Roto-Blade pump, this unit handles working pressures to 1,000 lb. per sq. in. continuous or 1,500 lb. per sq. in. intermittent. It is available as a single pump, double pump, single pump with feed pump, double pump with

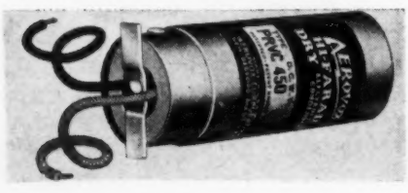


feed pump, and a single pump with gear pump. Capacities range from 8.5 to 50 gal. per min. Bearing loads are claimed to be neutralized by equalized and diametrically opposed radial hydraulic thrust loads. The impeller assembly, including the bushing and all moving parts, can be removed and replaced if necessary.

Capacitor

Aerovox Corp., New Bedford, Mass.

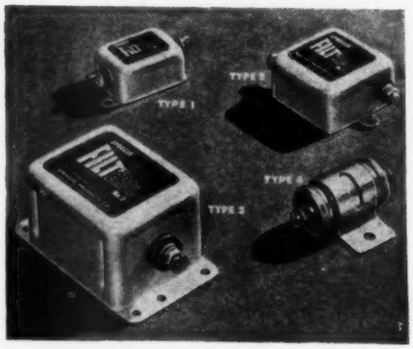
A general-purpose, cleat-mounting, electrolytic capacitor in an aluminum can. This unit provides a substitute for special metal-can capacitors requiring a mounting hole in the chassis or using the twist-prong mounting washer. The Type PRVC replaces twist-prong, spade-lug and screw-base types. These capacitors have insulated positive and negative wire leads, color-coded for polarity. Multiple-section units have concentrically wound sections with common negative lead.



Interference Filters

Sprague Products Co., North Adams, Mass.

These self-contained units are designed for installation in series with the power line or interfering device. Their basic circuit is a three terminal network of which the can is

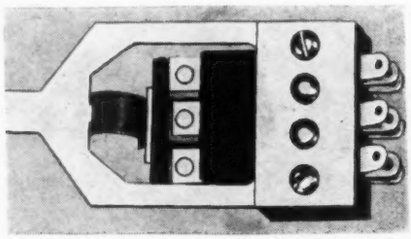


one terminal. Four available types include 115 volts a.c. or d.c. ratings from 1 to 35 amp. One unit for 220 volts a.c. or d.c. is rated at 20 amperes.

Snap-Action Switches

Acro Electric Company, 1310 Superior Ave., Cleveland 14, Ohio

Switches measure 5/16 in. in thickness and use operating pressures of 6 to 12 oz. Ratings are 15 amp. at 125 volts a.c. Available in fully inclosed models for vertical mounting with hinged leaf actuators and in models designed with mounting brackets combined with leaf actuators.



Ball-Bearing Turntables

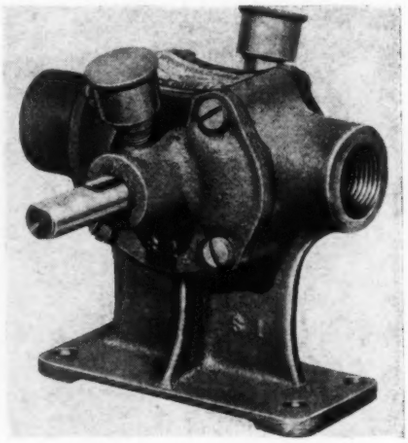
Whiting Corp., Harvey, Ill.

The turntable top (plain, grooved, raised track, or checkered) is carried on hardened steel balls, runnning in a machined ball-race. A foot-operated release lock, at table top level, holds table securely in position. The top overlaps bottom casting to prevent accumulation of dirt. Standard sizes range from 42 to 96 in. diameter.

Utility Power Pump

Jerome Simer Co., 422-DD Stinson Boulevard, Minneapolis 13, Minn.

The Simer paddle pump operates at 1750 r.p.m. (direct connected), and can be used at pressures to 30 lb. (70 ft. total head) including a 20 ft. suction. Will handle all chemicals except those that react on rubber, bronze or stainless steel.



Radio Interference Filters

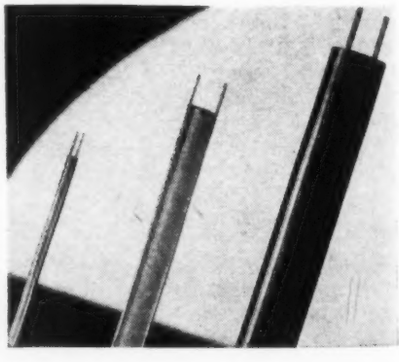
Solar Mfg. Corp., 285 Madison Ave., New York 17, N. Y.

These filters are of two basic electrical types, a balanced twin-pi filter for maximum attenuation, and a delta capacitor filter for normal usage. Known as Solar types EF-101 and EF-102, they are effective over the radio spectrum from 150 kilocycles to 150 megacycles.

Lead-In Wire

Federal Telephone and Radio Corp., Newark, N. J.

A weather resisting, low loss, polyethylene insulated twin conductor FM and television lead-in transmission line. These lines have



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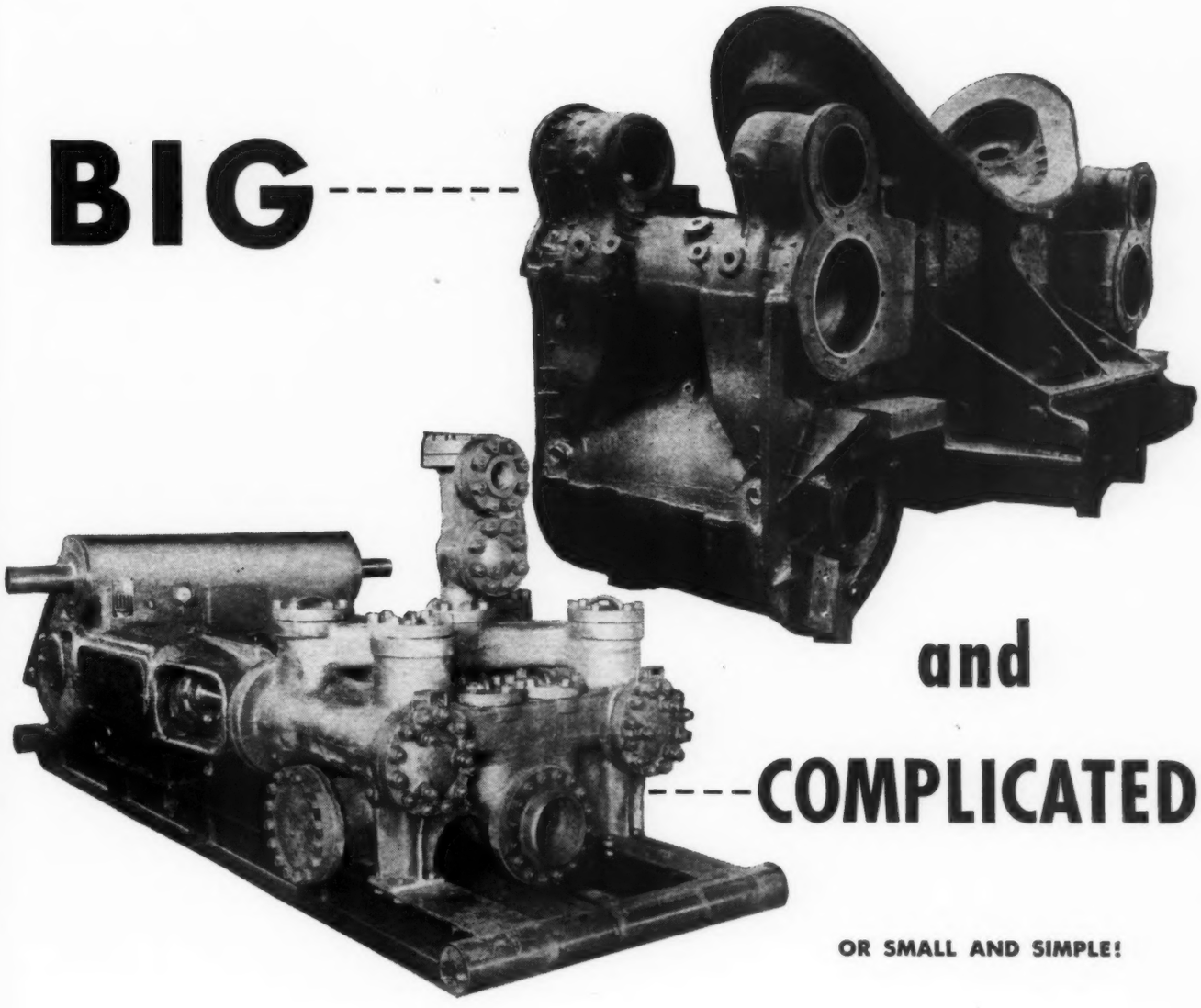

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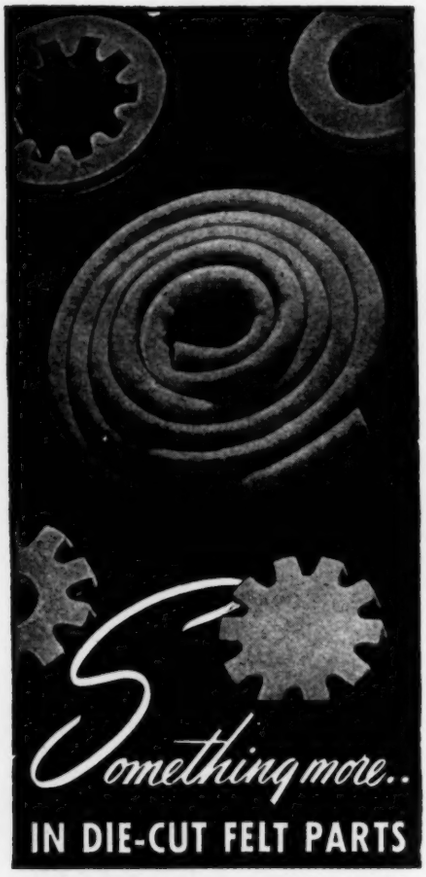

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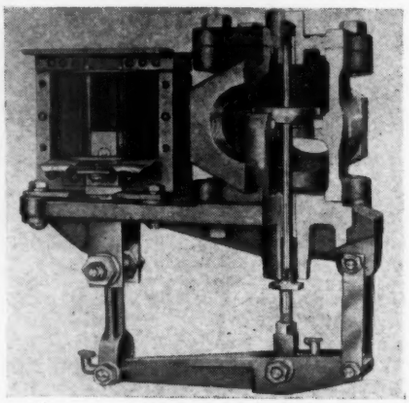
100, 200 or 300 ohm characteristic impedance. They are claimed to retain flexibility in sub-zero weather and the elliptical cross-section withstands twisting and abrasion.

Direct-Acting Solenoid Valve

Johnson Corp., Three Rivers, Mich.

Combining immediate full flow with the ability to operate under differential pressures to 150 lb., this valve can be used for all types of liquid level control, to 365 deg. F. The valve is available in two series. Type HH, sizes from ¼ to 1½ in. is suitable for differential pressures rang-

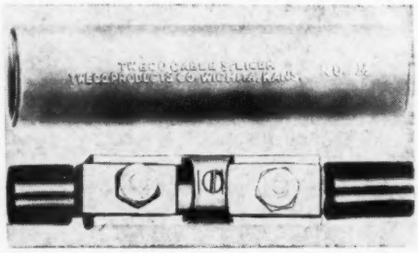


ing from 125 to 150 lbs per sq. in.; Type HL, sizes 1 to 3 in. is for medium and low differential pressures. Standard models are furnished for operation 110, 220, or 440 volt 60 cycle-current.

Welding Cable Splicer

Tweco Products Co., Wichita 1, Kan.

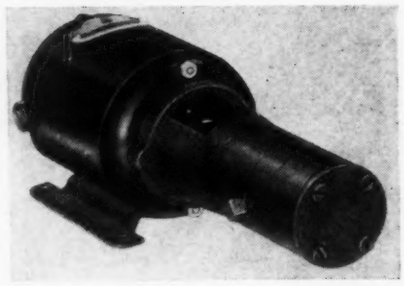
A clamp cable connection on each end of the splicer, with provision to solder between the cable ends makes the connection. A fibre sleeve covers the splice. Made in three sizes: Size "S" for small No. 4, 2, or 1 cables; size "M" for medium No. 1/0 or 2/0 cables; size "L" for large 3/0 and 4/0 cables.



Vane-Type Pump

Eastern Engineering Co., New Haven, Conn.

A positive pressure unit, Type V-W-1, designed for handling non-lubricating liquids. The pump has two composition bearings requiring no lubrication, and vanes made of the same material as the bearing. No centrifugal force is required to maintain volumetric efficiency. Pump delivery is approximately ½ gal. per min. The unit is



suitable for operating pressure of from zero to 30 lb. per sq. in. and is self-priming. The Universal motor is 1/15 h.p. and is available in 110 volts or 220 volts, a.c. or d.c.

Shielding Conduit

Chicago Metal Hose Corp., Maywood, Ill.

A synthetic-covered shielding conduit, known as RT-25. It consists of a galvanized steel flexible metal hose liner and synthetic cover. It is liquid-tight externally; is usable with both standard and water-tight conduit boxes; and is furnished with fittings.

Elapsed-Time Indicator

Marion Electrical Instrument Co., Manchester, N. H.

The Model HM3 indicator operates on 115 volts, 60 cycles and indicates elapsed time from zero to 9,999.9 hours. This instrument, hermetically sealed, is applicable for use where corrosive fumes or high humidity are present.